June 1941

MACHINE DESIGN

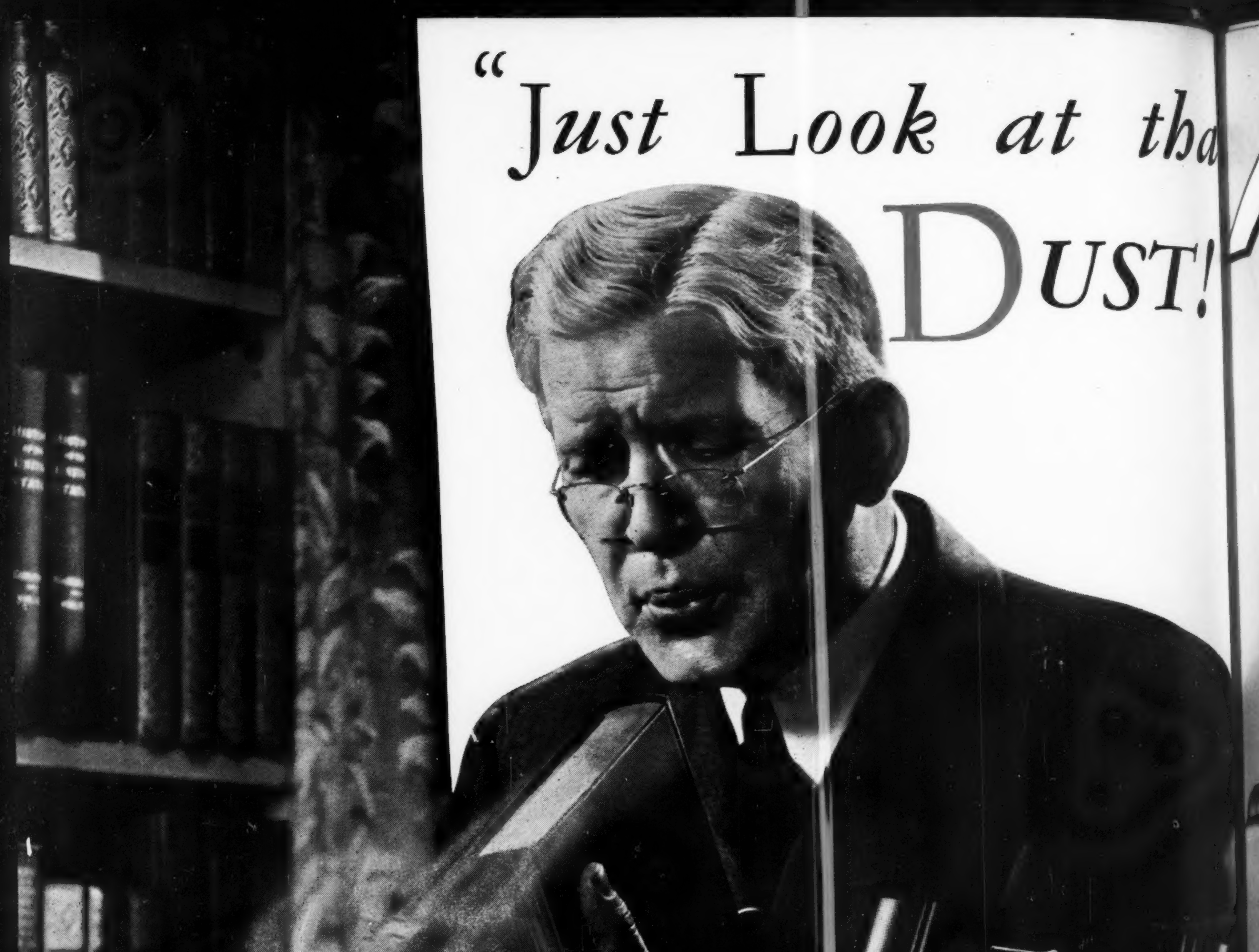
In This Issue:

Designing an Automatic Press

Springs under Static Loading

Checking High-Speed Motion

“Just Look at tha DUST!



Topics

INDICATIVE of the increased tempo of manufacturing operations are recent statements of the Machine Tool Builders, Timken, and Wright Aeronautical Corp. For the month of March, machine tool shipments are estimated at $57,400,000 as compared to about half that volume for March, 1940. Timken Roller Bearing Company's initial $3,000,000 defense program begun in 1940 will reach $5,000,000 by July. Also increased production is reflected by Wright's monthly output of 1,430,000 horsepower of aircraft engines. This is an increase of 23 per cent over January and 500 per cent over normal production.

♦

LOW cost, high dimensional stability, dry-process porcelain has been developed with mechanical and electrical properties equal to wet-process materials. It also is impervious to moisture and has resistance to shock equal to other porcelain molding compounds.

♦

SIMILAR to steel bonderizing, a new process been found to improve the affinity of aluminum for paint, lacquer and enamel coatings. In addition it protects the metal.

♦

QUESTIONS regarding extended uses of plastics resulting from the need to conserve strategic materials must be answered with: "Plastics are not panacea for the unfortunate deficiency in our defense materials." Plastics have many remarkable and successful applications, but as far as airplanes, for instance, are concerned, materials must be developed fully before applications can follow.

♦

ACTUAL flying conditions will be simulated for models of new military aircraft in a wind tunnel with a capacity for hurricane winds of 400 miles an hour. To produce this wind a 40,000-horsepower motor driving two 40-foot fans will be utilized.

PORTABLE ice rinks suitable for ice skating almost anywhere, even on a polished ballroom floor, have been developed recently. Using no piping system, the rink is composed of rectangular sections of brine circulating tanks and can be set up in a few hours. It is claimed that water sprayed on the sections produces a perfect skating surface.

♦

CIRCUMSCRIBED on all sides by obstacles, OPM's dollar-a-year men are in a delicate position. Carrying the chief responsibility for the success of the defense program, they deserve all the constructive help that engineers and industry as a whole can give them.

♦

REFERENCES to current literature and books on "subjects of pressing interest to those concerned with the technical aspects of national defense" are being compiled by the Library of Congress. The first bibliography has been issued, treating tanks and other military track-laying vehicles.

♦

EXTENDED use of porcelain enamel as well as plastics as discussed on Page 31 is expected to release strategic materials for defense purposes. Stainless and aluminum particularly can be replaced by enameled parts in refrigerators and other equipment.

♦

THE A.S.T.M. is developing new standard tests for plastic materials. These will be particularly helpful to those who are considering utilizing plastics to conserve strategic materials.

♦

TWICE the quantity of welding rods produced last year can be manufactured this year without difficulty, according to a recent statement of N.E.M.A. Thus 400,000,000 pounds of welding rod for 1941 would effect no "bottleneck" for manufacturers.

MACHINE DESIGN

Automatic Press Meets Current Conditions

By Stuart J. Myers
Design Engineer
Cropp Engineering Co.

BECAUSE of the comparative youth of the plastics industry and because a rapidly expanding market has left a margin of profit for even antiquated equipment, technological improvements have not been forced upon plastics manufacturers as rapidly as in other industries. By technological improvements is meant, more specifically, the elimination of direct labor as is normally accomplished in volume industries as a secondary stage in their development.

Marking a distinct advance over conventional compression molding presses, the press discussed in this article is fully automatic. The proper amount of molding powder is measured and placed in the mold, molding and curing operations are accurately controlled, the finished

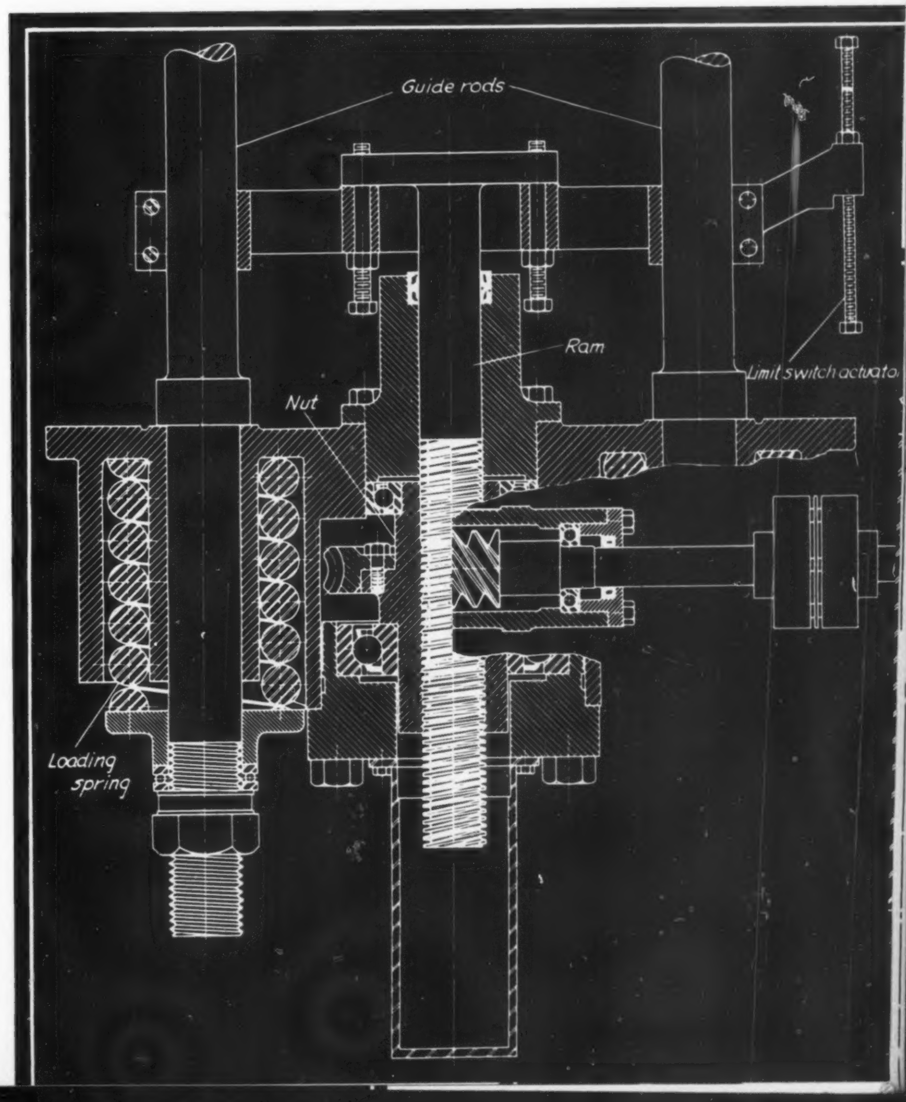


Fig. 1—Sectional view of worm and wheel ram drive shows location of loading springs in gearbox, away from heat

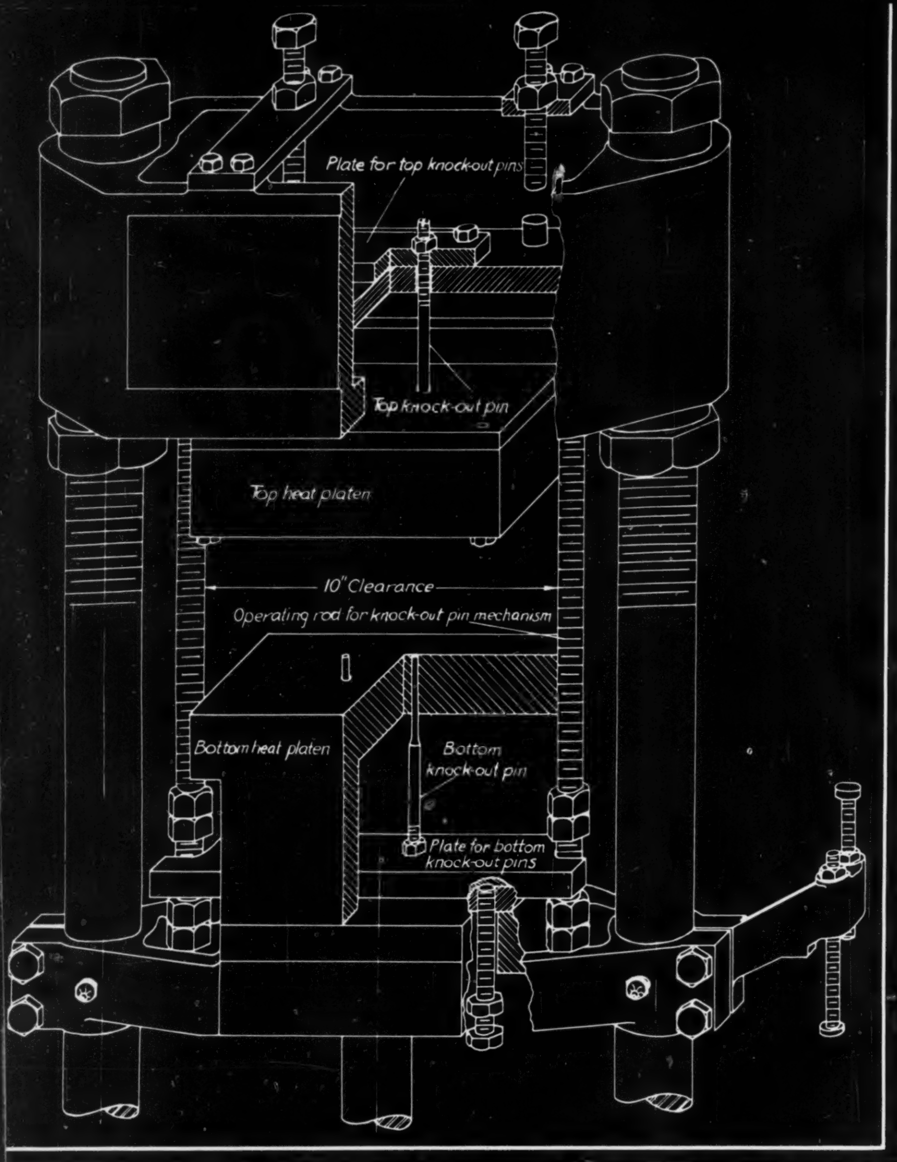


Fig. 2—Location and mode of operation of knockout pins are shown in isometric of press platen assembly

piece is removed from the mold and checked for defects, all without requiring direct operator attention unless a deflective piece is produced.

Despite the fact that the majority of molding presses are hydraulically operated, it was determined that for total pressures under 15 tons it would be desirable to obtain compactness and lower cost through completely mechanical operation. Several drive and power systems were considered in which a motor would be maintained in continuous operation and be made to build up a reservoir of energy which could be released for final pressures. However, after taking the matter up with motor manufacturers, it was learned that a five-horsepower motor could be built to special, high stalled-torque specifications which would allow 100 pound-feet of torque. Inasmuch as the pressure for curing can be held by spring pressure after once initiated, a high torque is only required for two-thirds of a second. Since the average requirement per cycle is only .8 horsepower when consideration was given the idle periods, the necessity for providing a flywheel or other energy reservoir was obviated, all energy for compression being taken directly from the motor.

Selection of the main driving motor having been accomplished, translation to the vertical, straight-line motion of the press platen presented a second problem. Consideration was given to toggles, cams, screws, and even racks and pinions. Toggles and cams were discarded because of the fact that the platen would necessarily have to close to the same point each time the press closed. This is undesirable when many different molds are used in the same press. The screw was chosen as preferable to the rack and pinion because of its self-locking feature and compactness of design in accommodating the high loads.

The final drive assembly is illustrated in *Fig.* 1. The platen ram or drive rod which is rigidly affixed to the platen is threaded to accommodate a nut. Any tendency toward rotation of this rod is opposed by the guide rods which pass through bushings in the platen.

In order to locate the motor at the side of the press where it would be out of the way, it was found necessary to use a worm drive so that the driveshaft would clear the springs used to maintain the pressure when the press was closed. By using a worm gear and a 12 to 1 ratio, triple-thread worm mounted on 4¾-inch centers an efficiency near 85 per cent was obtained. Buttress threads on the ram of four threads to the inch resulted in an overall efficiency of practi-

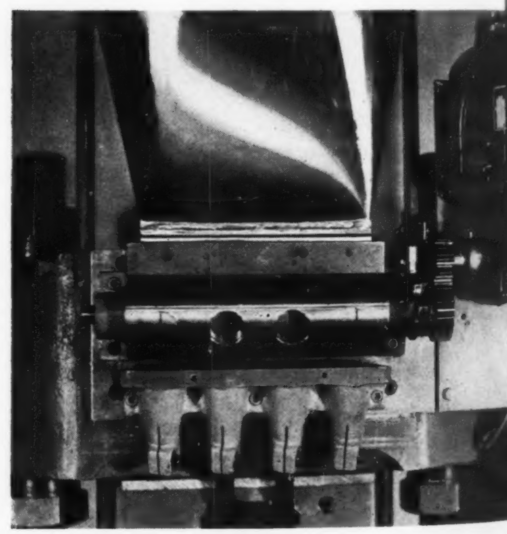


Fig. 3—Powder metering device with cover removed. Capacity may be adjusted by means of screw plugs

cally 15 per cent. Thus about 40 pound-feet of torque from the five-horsepower motor is used for about two-thirds of a second each time the press closes.

In order to insure long life of the buttress threads on the ram and in the driving nut, the nut was made six inches long so that, due to the large bearing surface in the engaged threads, the load

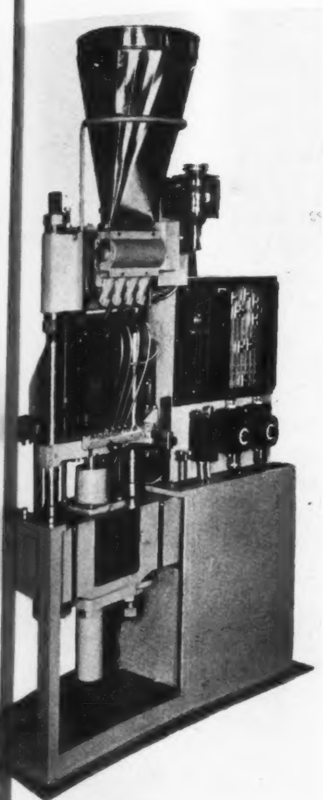


Fig. 4—Molding press produces plastic parts without requiring operator attention unless a faulty molding is produced

intensity on the threads was held to 884 pounds per square inch. To preclude still further the possibility of wear or seizure, the ram was made of a nickel-chromium alloy steel and the nut of a high tin bronze casting. The nut was turned to fit the worm gear and had a combination radial and thrust ball bearing on each end.

In addition to providing an ideal bearing combination with the bronze nut, the nickel-chromium steel used in the ram prevents distortion of the bearing surface of the threads. Also, because of its higher strength, a smaller diameter is possible. Consequently the helix angle is large, resulting in a screw efficiency of nearly 18 per cent. The ram is 2 inches in diameter, 4 threads to the inch.

Springs Isolated from Heat

The worm and its shaft are an integral forging carried on two radial-thrust bearings mounted in the worm housing. The worm housing is fitted to a wide groove in the side of the gearbox. The motor is connected direct to the worm shaft by a flexible coupling which allows the worm or any part of the gearbox to be removed.

In order to keep the gearbox as cool as possible it is placed below the mold. This also eliminates the possibility of oil dripping onto the mold and helps to keep the center of gravity down, as the motor and gearbox amount to one-third the total weight.

To keep the springs away from the heat, it was necessary to incorporate them in the gearbox. To do this and still keep the center distance of the side columns within reason, a special molybdenum cast iron alloy having a tensile strength of 50,000 pounds per square inch is used to withstand the transverse loading caused by mold pressure.

The head or top casting, shown in *Fig.* 2, must carry the top knock-out mechanism as well as the loading mechanism. As it is important that the knock-out pins be easily adjusted or removed, the top knock-out plate is placed inside the head but with the top of the head open for accessibility.

Loading mechanism proved to be the most difficult mechanical problem of the press. It is important that the loader be adjustable from 2 to 120 grams (about .2 of a cubic inch to 12 cubic inches) yet, at the same time, to be able to feed four mold cavities simultaneously. To do this the mechanism illustrated in *Figs.* 3 and 6 was developed. A high carbon, ground and polished shaft 3½ inches in diameter is mounted on two oilless bearings. The shaft has four radial holes each 1⅛ inches in diameter drilled almost through so as to form a cup-like cavity. These holes are threaded with a fine thread and have an adjustable bottom plug that varies the total capacity or amount of charge.

The top casting has an opening long enough to cover these holes and is machined to clear this shaft (or feed drum) by .005-inch. Under the shaft is a casting with four funnel-shaped cavities with a feed pipe from the bottom of each. Each pipe may be adjusted to feed its respective cavity in the mold.

Driven by a 1/10-horsepower gear reduction motor the shaft has a small cam integrally mounted on it which operates a switch that opens and stops the shaft in the proper position. The shaft can be removed for cleaning and changing powder color by removing four easily accessible screws.

If a plastic part, in being extracted or "knocked out" of the mold, were to break in such a manner

Fig. 5—Above—Timer bar or trolley permits accurate sequence control of press operations

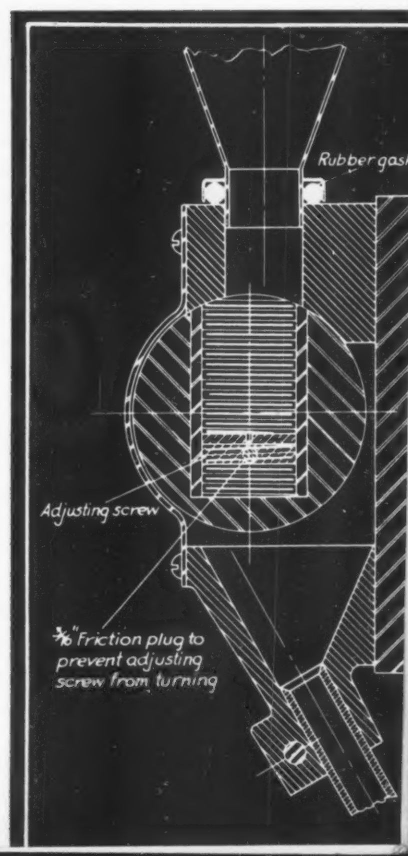


Fig. 6—Right—Sectional end view of metering device shown in Fig. 3

that a fragment remained in the mold, successive pieces would also be imperfect. To insure against such a contingency, two safeguards are provided. First, after the knock-out pins have removed the piece, air jets thoroughly clean the mold. Second, these same air jets blow the piece onto a sensitive balance which operates an electric contact. If the piece is broken, and hence a part is missing, it will be too light to trip the contact and the press will be stopped. To avoid the possibility of even a light piece tripping the balance upon impact with the platform, an air jet is directed against the underside of the platform while the part is falling upon it.

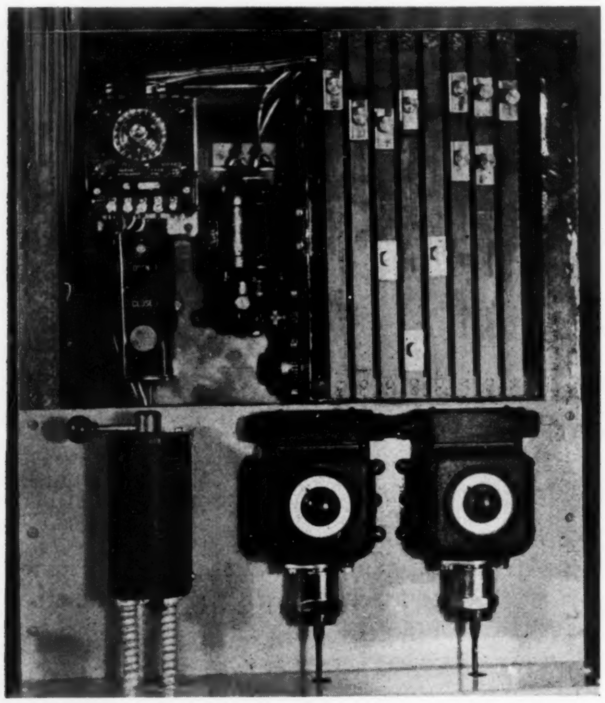


Fig. 7—Control panel shows eight timer bars in position. Control in upper left adjusts curing time from 2 to 20 minutes

Fully automatic operation of the aforementioned functions of which this machine is capable necessitates a precise control system. The following operations must take place in proper sequence and after the proper time interval:

1. Filling of the cavities in the loading drum illustrated in *Fig.* 3
2. Emptying of the powder into the mold
3. Closing of the press
4. Opening of the press
5. Stopping of ram
6. Turning on of the compressed air to remove the finished piece from the platen and cleaning the mold
7. Turning off of the air.

In addition to the opening of the press indicated in operation 4, it is sometimes necessary to open the press slightly during the curing time to "breathe" or "de-gas" the mold. This operation is also provided automatically.

Because of its simplicity and efficacy, the design of the control mechanism merits detailed consideration. All of the control circuits are normally open so that any of several parallel switches can effect an operation by being closed. In one instance, this meant the incorporation of an extra relay but resultant simplification of the mechanical design of the timer proved ample justification. Snap-action switches are mounted with a yoke so that they can be slid up or down a trolley or timer bar in which two brass tracks in a bakelite strip continually make contact with the switch regardless of its position. A thumbscrew allows the switch to be moved in any position with extreme accuracy and throughout the entire length of the cycle. A back and front view of one of these timer bars is shown in *Fig.* 5. The bars are mounted close together and vertically as shown in *Fig.* 7 so that if a horizontal rod were moved from the top to the bottom it would press the actuating points of each switch, thus closing its circuit. This vertically moving horizontal rod is fastened between two roller chains which are driven by a 1/50-horsepower gearmotor.

Springs Maintain Mold Pressure

The first two controller bars are used to control the loader. Number one determines the time at which the loader is filled, and number two controls the part of the cycle at which the loader delivers its powder to the cavities. The vertical distance between the switches on these two adjacent bars determines the time interval between the two operations.

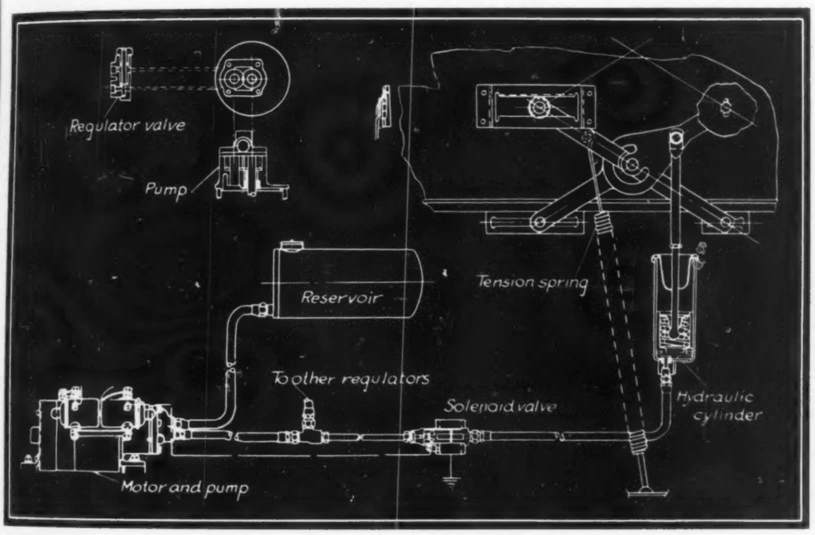
The switch on the third bar, when closed, energizes the contactor which starts the motor and closes the press. When the bottom ram has compressed the springs to a point where sufficient pressure is applied to mold the piece in question, an electrical limit switch actuator, shown in *Fig.* 1, opens the interlock on the closing contactor and thus stops the motor. The amount of pressure applied to the molds is a direct function of the compression of the springs, and the location of the limit switch is adjusted by the calibrated knob so that any pressure between two and twelve tons is available at fingertip control.

Automatic Control "Breathes" Mold

The next bar or circuit is one which, when closed, actuates the opening contactor which opens the press. As shown in *Fig.* 7 two parallel switches are mounted on this bar inasmuch as on certain types of moldings it is desirable to "breathe" or "de-gas" the mold for an instant. After this degasing period is over the mold is again closed and the timer is actuated, which times the period to a specific number of seconds of curing. During this

(*Concluded on Page* 120)

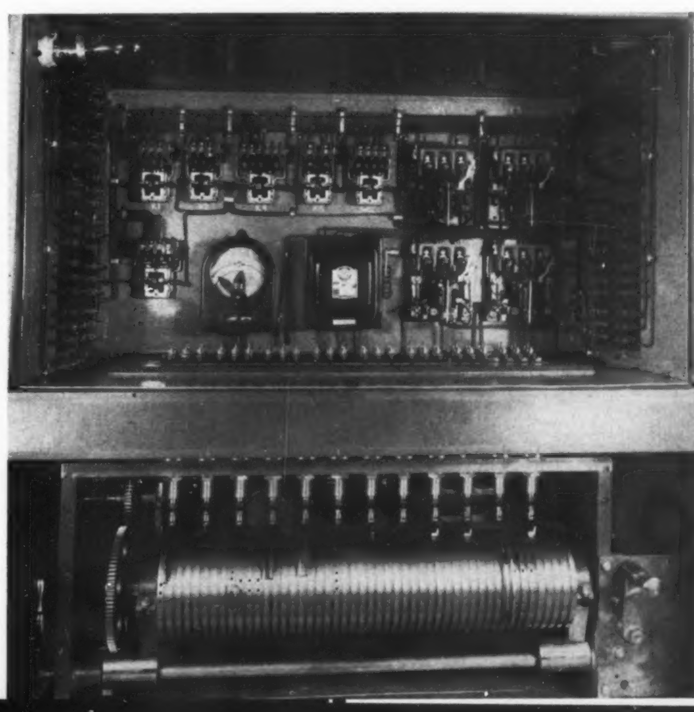
Scanning the Field FOR IDEAS

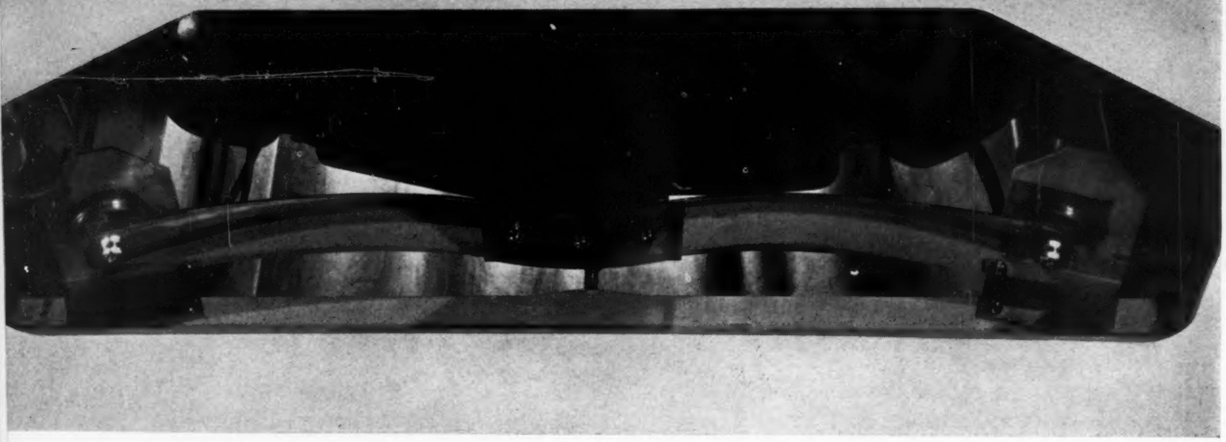


Convenience is the keynote in the refinement on Chrysler cars with power-operated windows. Hydraulically motivated as shown in the accompanying sketch, left, each window is operated by a hydraulic cylinder controlled by pushbutton on the dash. In each pressure line leading to a window is a solenoid valve to regulate the movement of the hydraulic fluid. Additional pushbutton control stations are also located near the windows in the rear.

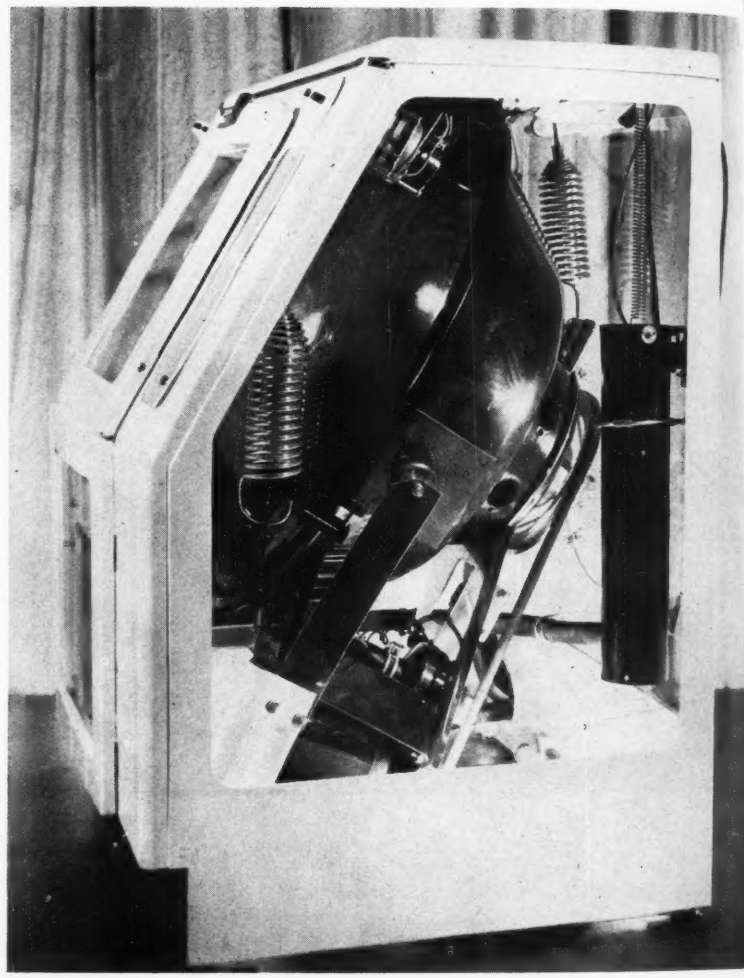
Hydraulic power is supplied from a pump driven by a motor with four-field coil design, wound for reversing duty to operate the pump in either direction. Pump is a conventional gear pump with spiral gears to eliminate noise. Operation continues as long as button is depressed or until window reaches its limit of travel. This feature allows adjustment of all the windows, up or down, to the various levels desired.

Sequence control and accurate timing of operations are accomplished by this drum with prepositioned cams as shown in the illustration at right. Actually the "brains" of a Budd unit for induction hardening internal surfaces and cylinder bores, all of the machine sequences are maintained by this controller which consists of a series of precision cam-operated switches. The drum is arranged to receive four sets of cams, the different sets controlling the operations for different sizes of cylinders to be treated.

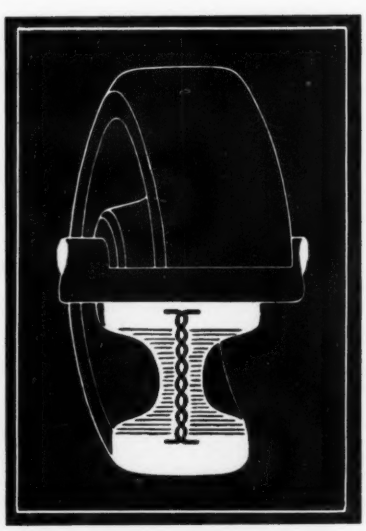


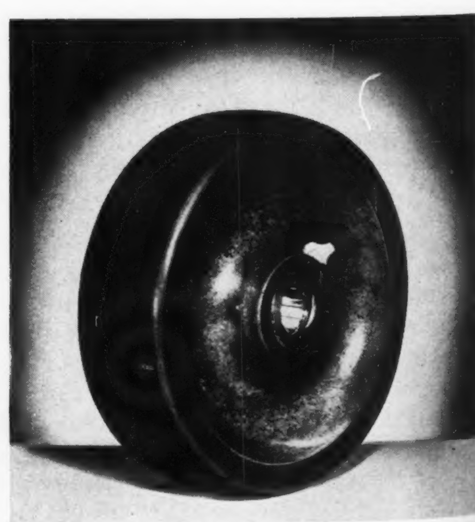


Vibrations are absorbed and snubbed to the extent that no noticeable disturbances are transmitted from the whirling tub to the cabinet of this Westinghouse washer. The suspension system utilizes triangulated coil springs, two in front and one in back of the rotating cylinder. The springs cushion the vibrating forces which in turn are smothered by five snubbers—two above, one on each side, and one below the suspended unit. Arrangement of the suspension is shown in the cutaway illustrations above and at right.



Electrical conductor is molded into laminated-plastic truck wheels to ground the bushing or hub to the rim as shown in the drawing and photograph below. Formica plastic, the molded rim contains colloidal graphite filler which is an excellent conductor. Thus a truck on which these wheels are mounted is grounded through wheel hub, flexible conductor and conducting rim. The plastic wheels, also, will not spark upon striking an abrasive, making the trucks especially suitable for handling powder or highly combustible distillates.





Stopping Motion for Design Data

By Kenneth D. Moslander

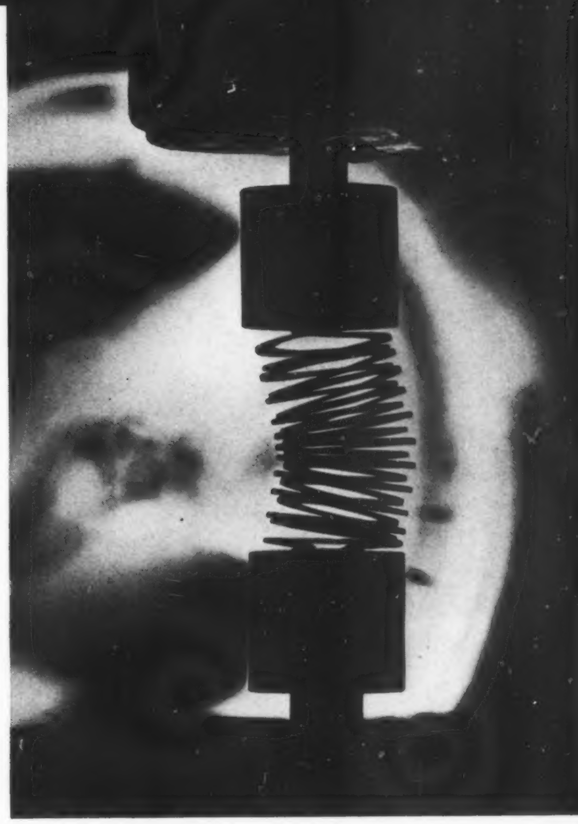


Fig. 1—Interference between the two concentric helical springs dictated the impracticability of this type shaft coupling for high speeds

APPLICATIONS of high speed camera and stroboscopic equipment have been confined so largely to speed measurement and trick pictures of bullets leaving a rifle that its value as a tool of the machine designer has been, to some extent, neglected. Even where high-speed investigation has for its purpose the obtaining of design data, the possibility of extending its use to other than simple rapidly rotating or reciprocating machine parts is often overlooked. Case histories, presented herein, have for their purpose the acquaintance of the design engineer with the broad diversification of high speed investigations possible with standard equipment plus a modicum of native ingenuity.

Two basic techniques are in use for the investigation of high-speed motion. The first of these is by use of special motion picture equipment; the second, with stroboscopic light sources with or without the adjunct of a "still" camera.

High-speed motion picture photography, both from the standpoint of equipment and operation is far more costly than the simpler and comparatively inexpensive stroboscopic method. Whereas in certain non-periodic phenomena such as progressive chemical reactions the motion picture camera is indispensable, for the major number of machine investigations the stroboscope is more than adequate.

When, for the purposes of record or protracted study, photographs of "stopped" high-speed motion are desired, an auxiliary light source synchronized with the stroboscope is used. Such a unit provides a light source 100 times as intense as that of the primary stroboscopic flash. Since the duration of the flash is in the order of ten-millionths of a second, when the flash frequency is adjusted to the speed of the machine part under investigation, time exposures of sufficient length to produce satisfactory results may be made with practically any kind of camera.

One such photograph is shown in *Fig.* 1, taken in conjunction with the development of a special shaft coupling. This development was primarily con-

Fig. 2—Right—Distortion of strip spring loops observed by the stroboscope, resulted in excessive variations of end thrust on shaft

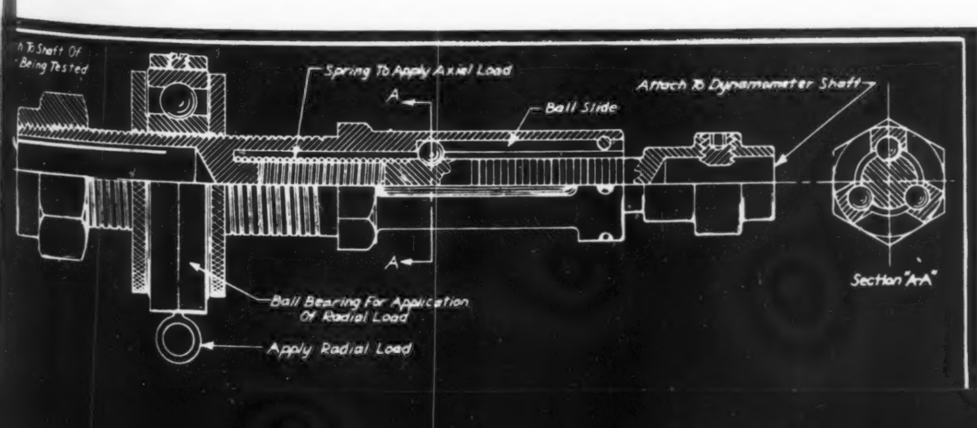


Fig. 3—Left—Final design of coupling permits fine adjustments of end thrust and radial load

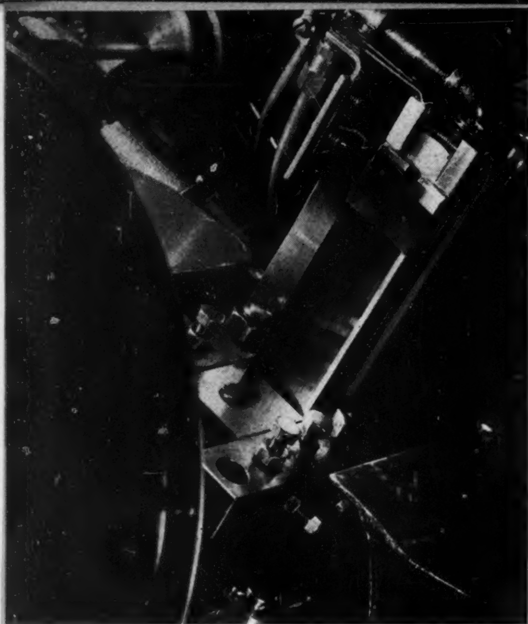


Fig. 4 — Data on piston ring and cylinder bearing phenomena indicates burnishing may be a prelude to failure

cerned with providing means for connecting a motor to an electromechanical type dynamometer in such a manner that both axial and radial loads could be applied to the shaft of the motor under test.

Specific requirements were that the coupling must transmit torques produced by motor having outputs up to .4-horsepower at speeds of from 5,000 to 14,000 revolutions per minute. The coupling was also to provide an axial load on the motor shaft capable of being adjusted from zero to 60 ounces.

To eliminate friction losses which would be the inevitable consequence of sliding parts, a coupling of the type illustrated in *Fig.* 2 was tried. Stroboscopic examination disclosed that the leaf springs were so distorted under the influence of centrifugal effect as to produce wide variations in the axial load on the shaft.

Helical Springs Investigated

A coupling utilizing multiple helical springs was then investigated. Effect of centrifugal force was thus reduced to within practical limits but, as shown in *Fig.* 1, interference between the springs in the coupling when rotating at high speed resulted in uncontrollable variations in operating characteristics.

Shown in *Fig.* 3 is the finally developed coupling. Evident from the foregoing is the extent to which the stroboscopic investigation contributed to the success of the perfected part.

Several extremely interesting conclusions relevant to bearing characteristics of reciprocating parts were arrived at by means of the stroboscopic set-up shown in *Fig.* 4. A section of an engine cylinder was so mounted that it could be reciprocated at a rate of 620 cycles per minute. Applied to this cylindrical surface is a segment of a piston ring in which is embedded a thermocouple for measuring bearing surface temperatures.

To make possible a stroboscopic investigation of the surfaces and to observe the progress of burnishing and scuffing, a limit switch (shown in the upper right of *Fig.* 4) is used. This switch actuates the flash lamp, thereby effectively "stopping" the motion for visual or photographic observation.

Initial loading of 600 pounds per square inch is increased at the rate of 100 pounds per square inch every five minutes until failure occurs. On the basis of data obtained from these tests it was concluded that bearing failure follows shortly after the surface acquires a high polish or burnish. This burnished surface is not as good a bearing as the rougher surface which immediately preceded it. In fact, the highly polished surface can never again sustain as high a load as it did while acquiring its ultra-smooth surface.

Due to the fact that a rougher bearing surface has less actual contact area, measured temperatures are higher. However, actual load-carrying capacity for failure is also higher. Indications were also obtained that a rougher surface constitutes a better bearing because of the property of the surface to keep the slight irregularities at a lower temperature due to better cooling rather than because a surface "seats" more quickly or has better oil-carrying capacity.

By means of this and similar investigations utilizing stroboscopic equipment, materials and surface treatments have been developed for piston rings which result in their withstanding much more severe operating conditions than were previously possible.

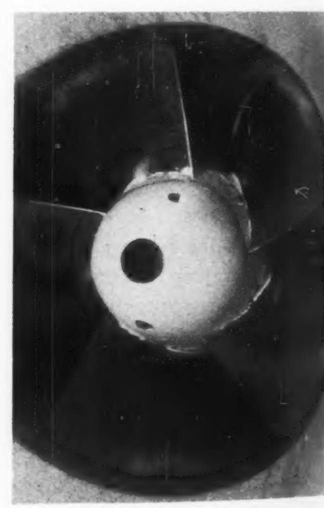


Fig. 5—Cloudy cavitation areas indicate imminent pitting of the blades and decrease in overall efficiency

Cavitation phenomena in hydraulic turbine runners, depending, as it does, upon such widely divergent influences as the number of turbine blades, blade pitch, speed, head, power output, efficiency, etc., does not lend itself easily to analytical investigation. Yet cavitation is more responsible than any other single factor for unsatisfactory operation of some turbines. It has, as its direct effect, severe pitting of the runner plates. This effect is combatted by applying a welded deposit of stainless steel to those sections of the runner likely to be affected.

Until the application of stroboscopic equipment to this problem, the location of these stainless steel reinforced areas was largely a rule-of-thumb procedure based upon an extensive background of experience as well as protracted tests on models.

Use of the stroboscope and camera in this investigation was compassed by seemingly insurmountable difficulties. First was the problem of providing a transparent window in the turbine housing which would blend with its contours and not deflect under pressure with consequent introduction of new variables. Second, it was necessary to obtain a sufficiently intense light source so that the beam, after traversing the window, several inches of water, the cloudy cavitation areas and return, would still be able to produce a usable image on the camera film. Third, because of the localized low pres-

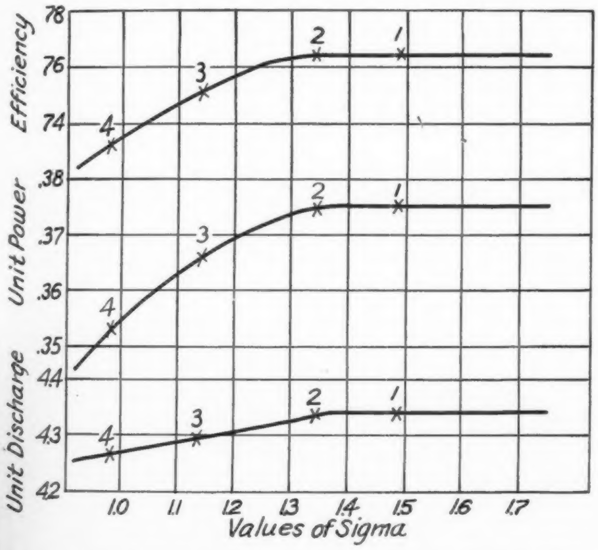


Fig. 6—Stroboscopic investigation reveals that cavitation may occur in localized areas well out on the flat portion of the sigma curves

sures experienced (vacuums as high as 25 inches of mercury are not unusual) dissolved air comes out of solution producing cloudiness in the water which makes photography impossible.

In the face of such obstacles, it is not so remarkable that usable photographs such as *Fig.* 5 were obtained as that photographs were obtained at all. A transparent acrylic resin window was built into the model housing. An outer glass window was installed and the chamber between connected to the draft tube water passage. Thus, pressure was equalized on both sides of the plastic window.

Light intensity from the standard stroboscopic flash bulb was increased by adding supplementary condensers to the circuit. Under these conditions continuous operation of the bulb was impractical but intermittent single flashes were obtained which proved entirely satisfactory.

Cloudiness resulting from entrained air was overcome by subjecting the entire system to relatively high pressures at which cavitation does not occur. The pressure is then rapidly reduced to the desired point of investigation and a photograph taken with a single flash before cloudiness in the water becomes too severe. Evidences of cavitation are clearly shown in the photograph, *Fig.* 5, taken in this manner. The entire system is maintained in total darkness; the camera shutter is held open until the single stroboscopic flash occurs.

Knowledge of Cavitation Extended

Prior to the investigation of cavitation phenomena with the stroboscope it was generally assumed that cavitation started at the break in the sigma curve, *Fig.* 6. In other words, that the incidence of cavitation in a turbine runner was identical with the point at which efficiency begins to fall off. Actually the stroboscope revealed localized cavitation areas well out along the flat portion of the curves.

In the automotive industry many occasions are found for the use of the stroboscope. One of these is illustrated in *Fig.* 7. In this case it was desired to analyze chassis and body movements under rough road operating conditions. The automobile rests on rollers to which metal strips are fastened to simulate the effect of a "wash-board" road. By synchronizing the light flash with the rotating roll-

Fig. 7—Stroboscope is as effective in indicating the essential lack of motion as in measuring frequency and amplitude of periodic vibrations

ers, periodicity and amplitude of vibration of all parts can be analyzed and the various phase relationships established.

In the investigation of fluid couplings, transparent housings were built through which the motion of the oil could be studied. The stroboscope provides, also, an interesting means of measuring slip. By synchronizing the flash to the speed of the driv-

Fig. 8—All guess-work is eliminated from the design of propeller counterweights by observing their action at high speeds with aid of stroboscope

ing member and then counting the number of apparent revolutions per unit time of the driven member under the stroboscopic light, direct, quantitative slippage data is obtained.

Manifold applications are found in the study of gasoline engines. Behavior of valve springs under high-speed conditions can be dealt with satisfactorily in no other way. In one such instance in the course of a development of a 7000 revolutions per minute racing engine, it was necessary to construct an analagous test set-up utilizing a direct current motor. Contacts mounted on the end of the test camshaft afforded actuation of the flash lamp and permitted a detailed study of spring surge as well as a quantitative determination of spring loads necessary to maintain proper following of the cam. By making the stroboscopic commutator on the end of the camshaft adjustable through 360 degrees, examination of the spring action at any phase of the valve opening is possible.

Propeller Balancing Improved

In the test set-up illustrated in *Fig.* 8 pertinent data was obtained on the dynamic balancing of propellers. Observation of the slippage of the balancing weights made possible by the stroboscope led to a redesign which eliminated the difficulty.

In other phases of aircraft development the stroboscope plays an equally important part. Vibration studies of fuel and oil lines, elimination of resonance frequencies in radio equipment, effect of variations in engine speed on the several air foil sections, are but a few of the uses to which the stroboscope has been successfully applied.

Illustrating an unusual use of the stroboscope in conjunction with a conventional motion picture camera, an investigation of inertia effects was conducted on a needle which carries the filling yarn in an Axminster loom. This needle is about five feet long and, while it moves at high velocity, its speed in cycles per second is not sufficiently high for stroboscopic investigation. However, for design purposes it was necessary to obtain only displacement, velocity and acceleration data which could be measured while the needle was passing a point.

To accomplish this the set-up shown in *Fig.* 9 was used. The needle was calibrated in equal increments throughout its length. A slot was made in the casing upon which was focused both the flash bulb and the camera. By synchronizing the stroboscope to the camera by means of a commutator on the camera shaft and operating the latter at 64 frames per second, pictures were obtained in which

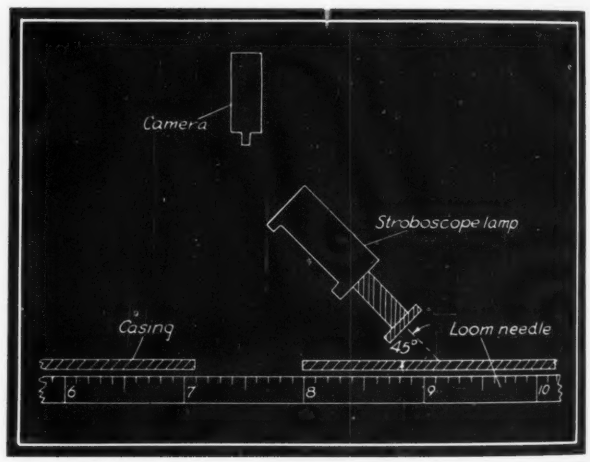


Fig. 9—Displacement, velocity and acceleration data obtained by motion picture camera and flash bulb resulted in design improvement of loom needle

the graduated marks on the needle could be read.

It was thus a simple matter to plot graphically the displacement, velocity and acceleration data, obtaining a clear picture of the inertia forces in the needle itself as well as the loads in the needle drive mechanism. Since the time interval between each frame of the motion picture film is 1/64-second, the difference between the graduated scale readings, as observed in adjacent frames, is equal to the distance traversed in that time.

MACHINE DESIGN gratefully acknowledges the assistance of the following companies in the preparation of this article: American Airlines, Inc., *Fig.* 9; Baldwin Southwark Division of Baldwin Locomotive works, *Figs.* 6 and 7; Bigelow-Sanford Carpet Co. Inc., *Fig.* 10; General Radio Co., Gulf Research and Development Co., The Hoover Co., *Figs.* 1, 2 and 3; the Perfect Circle Co., *Fig.* 4.

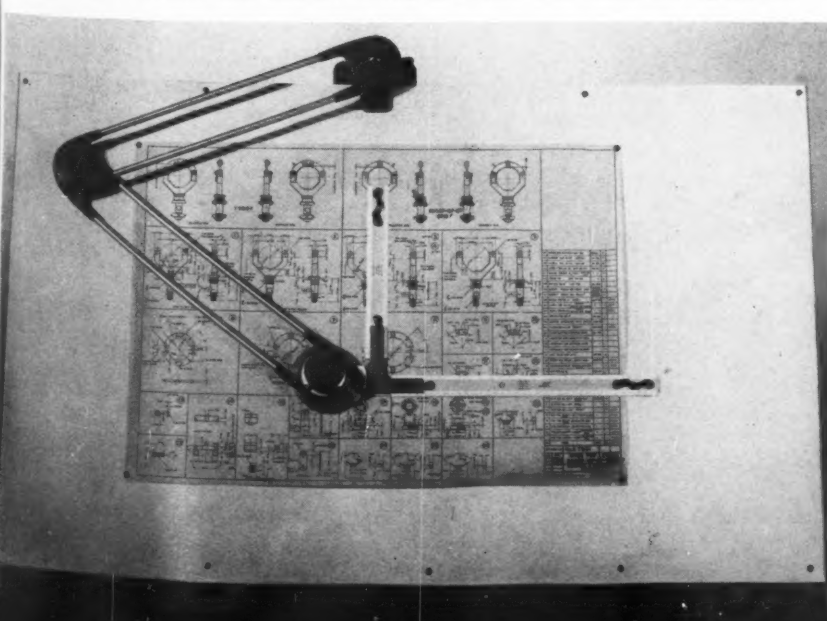
Specifying DESIGN DEPARTMENT MATERIEL

Part III — Special Equipment

By H. T. Pentecost

SPEED and utility are the yardsticks by which, under the press of present conditions, the value of engineering department equipment must be measured. A machine part evolves from an idea to a drawing, to a reproduction, to the shop and finally to the finished part. This third article in the series concerns itself with the second step in the evolution, the production of the drawing and the care necessary in its execution.

Fig. 1—Drafting machine eliminates the necessity of T-squares, triangles, protractors and scales; it requires but one hand for operation



Attributed to O. P. M. administrator Mr. Knudsen is the statement that a battleship requires, in the making, three tons of blueprints. The startling significance of such a number lies in the hours, days and weeks of drafting time taken to produce the drawings behind these prints.

Equal in importance to speed is precision and clarity of detail. The acute shortage of skilled machinists in most shops today, makes it imperative that the specifications for a part be such as to be readily comprehended by the semiskilled machinist who will probably do the job.

The one-armed paper hanger is put to shame by the dexterity with which a draftsman may manipulate a T-square, a couple of triangles

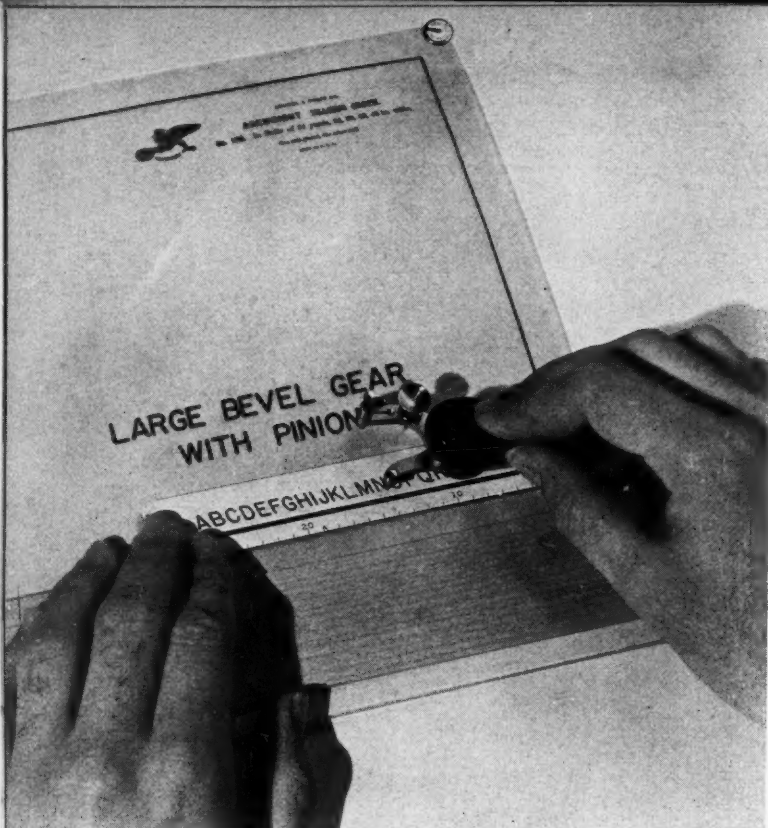


Fig. 2—Hand lettering equipment assures uniformity of perfect letter forms on all drawings

and a pencil. However, the final measure of his efficiency is not his expenditure of energy but the number of accurate lines he puts on his drawing. More than by any other single factor, his speed and precision are enhanced by the type of drafting machine illustrated in *Fig.* 1. Reducing drafting time by as much as 40 per cent, such a machine replaces T-squares, triangles, scales and protractors. Every part of the design in the board is readily accessible for the drawing and measuring of lines and angles; perfect parallelism is assured at all times. Since all lines are drawn along the straight edge of a graduated scale, their length is correct the first time, obviating the necessity of redrawing or erasing ends.

Lettering is one of the most tedious and time-consuming jobs the draftsman is called upon to do. Whereas many can consistently produce character forms of almost machine-like precision, some are completely unsuccessful in developing this art. In any event, however rigorous are the departmental specifications as to size and style, the maker of a drawing can usually be identified by his characteristic lettering.

All such difficulties are effectively overcome by the use of the lettering instrument illustrated in *Fig.* 2. Working from a selected templet containing the desired size of characters, the draftsman can make either vertical or slanted letters by a simple adjustment of the pen arm. Thus complete uniformity of lettering is achieved throughout the department while, at the same time, speed is materially increased.

Letters at High Speed

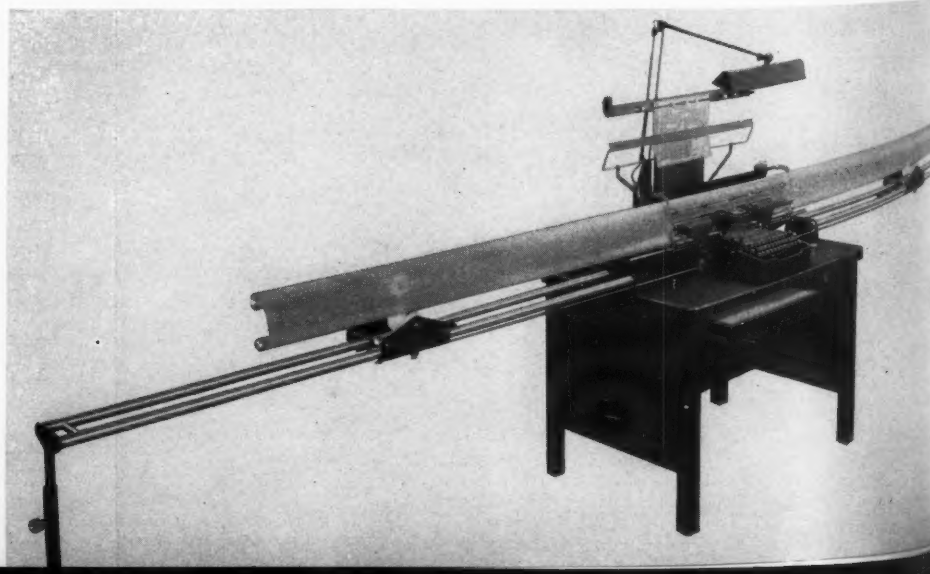
For large engineering departments where the quantity of lettering on drawings justifies its use, the machine illustrated in *Fig.* 3 should prove of incalculable advantage. Capable of printing in a manner indistinguishable from the best of hand lettering, the machine operates like a typewriter and will letter at typewriting speeds. Either cloth or paper drawings or tracings up to 12 feet in width and of any length are handled easily. A wide variety of letter styles are available in sizes from 6 to 14 point, more than covering the ordinary range of sizes used on engineering drawings.

Since a maximum size drawing of 12 feet in width can be shifted from one end to the other in a few seconds, a complete lettering job can be accomplished with a claimed saving of from 30 to 60 per cent. The possibilities of this machine are apparent. A stenographer can, from notes, do all the lettering on engineering drawings, freeing the draftsman or designer for the more important work of developing the designs. In the light of the present shortage of skilled men this should be a material consideration.

Solves Filing Problem

After a drawing is completed, the problems of the engineering department are by no means over. Rarely is a drawing destroyed; they are preserved in files down through the years, becoming yellow and brittle with age but taking up just as much space as they ever did. Thus filing space requirements for engineering drawings becomes a direct

Fig. 3—Operated like a typewriter, the machine at right simulates hand lettering on drawings 12 feet in length



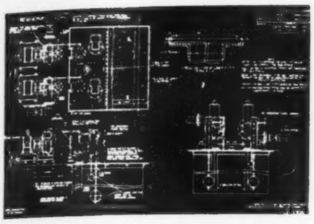


Fig. 4—Actual size film negative made from a 20 x 30-inch engineering drawing

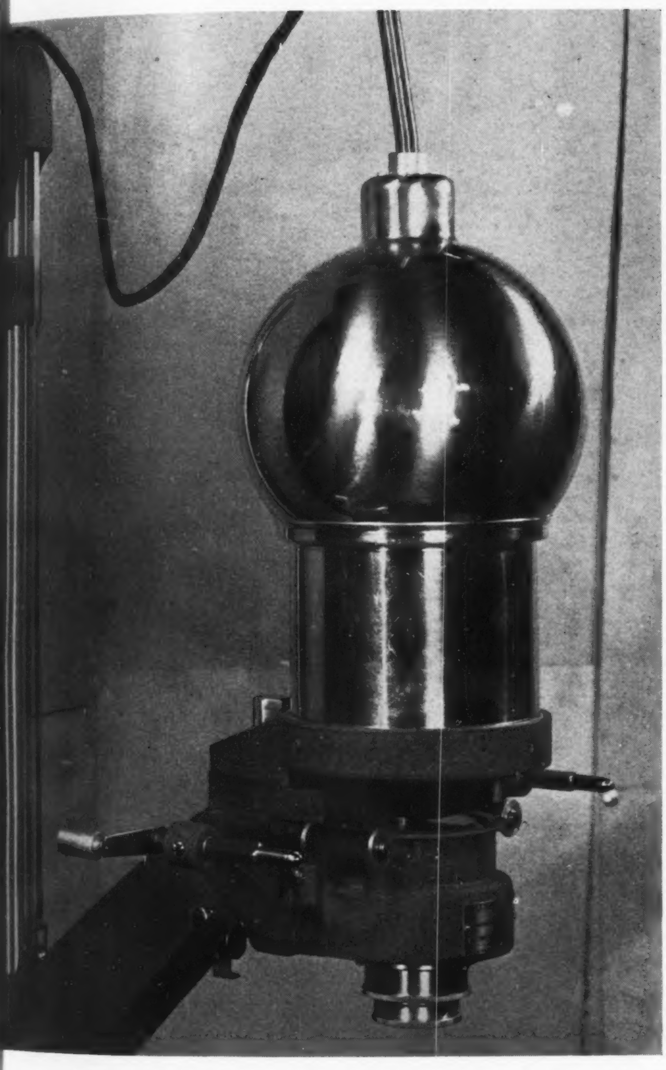


Fig. 5—Enlarger will make full-size tracings, capable of being reproduced by conventional methods, from negative photographs on 35-millimeter film

function of the product of the production rate and the number of years the department has been in existence.

With the aid of the equipment illustrated in *Fig.* 6 these difficulties are overcome. Capable of making sharp, accurate reproductions on 35-millimeter film of drawings up to 37½ x 52½ inches, the machine will free a large portion of the space now occupied by files for more productive work. Files of these films require only about 2 per cent of the space needed by the drawings replaced. Shown in actual size in *Fig.* 4 is an illustration of a 20 x 30-

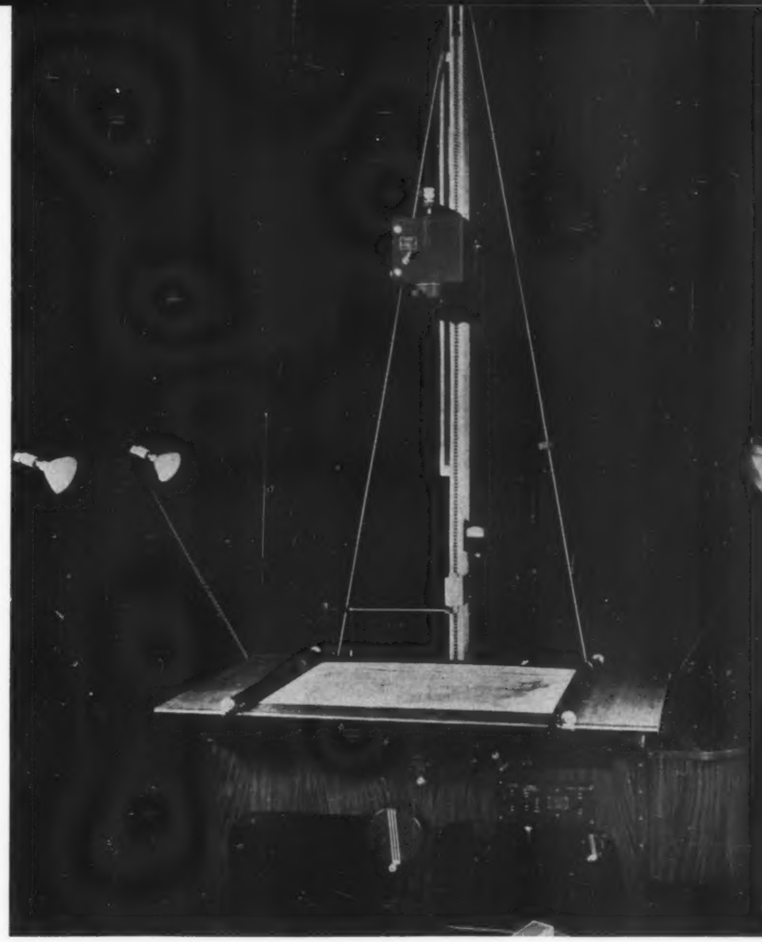


Fig. 6—Photographic equipment is capable of preserving drawings up to 37½ x 52½ inches on 35-millimeter film in greatly reduced space

inch drawing as it looks after having been photographed. From 450 to 500 drawings can be photographed in one hour, 100 feet of film accommodating about 1100 18 x 24-inch drawings. It is thus evident that one machine can not only easily keep pace with regular production but, at the same time, convert existing drawings to the film system.

Film Facilitates Reference

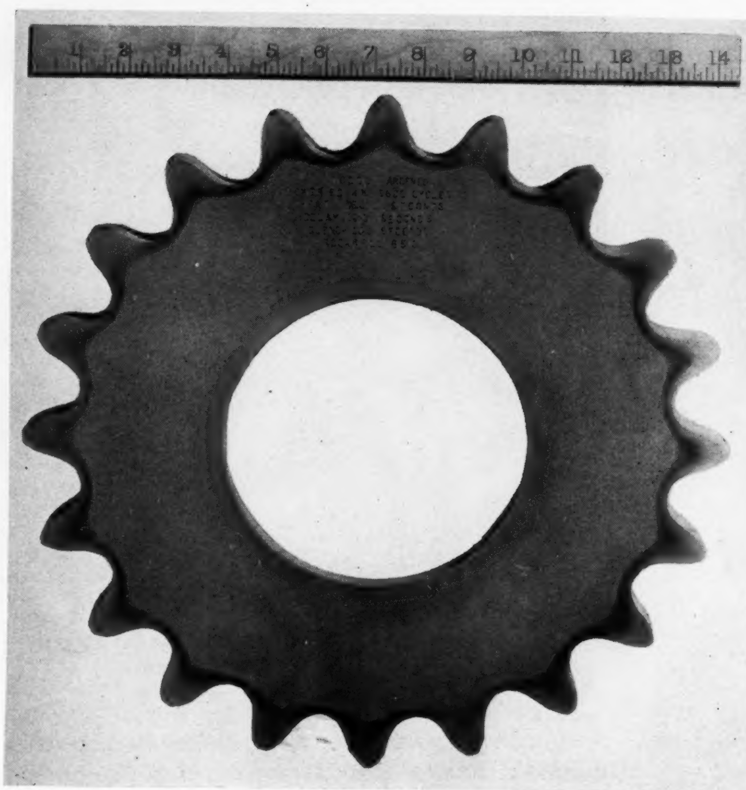
Reference to these filmed drawings may be had by the simple expedient of projecting them upon a ground glass field. The clarity of the original drawing is preserved indefinitely. When shop prints are desired from drawings in the film file, the original drawing can be easily and quickly reproduced to original size on sensitized tracing paper or cloth from which conventional reproduction prints can be made. Enlarging equipment designed primarily for this work is shown in *Fig.* 5.

A further apparent advantage of this system is that a complete set of detail drawings can be kept, if desired, on a single length of film. By so doing, the risk of loss or misplacement of any one is obviated.

MACHINE DESIGN acknowledges with appreciation the assistance of the following companies in the preparation of this article: Keuffel & Esser Co., *Figs.* 1 and 2; Ralph C. Coxhead Corp., *Fig.* 3; Recordak Corp., Subsidiary of Eastman Kodak Co., *Figs.* 4, 5, and 6; The Frederick Post Co.

Fig. 1—Right—4-inch crankpin bearing hardened to 55 rockwell C. Depth of case is ¼-inch

Fig. 2—Sprocket demonstrates the hardness distribution obtainable in irregularly shaped machine parts



Induction Hen

Steps Up uc

INDUCTION hardening makes possible high production of locally hardened machine parts with the desired degree and depth of hardness, with metallurgical structure of core essentially unchanged and with practically no distortion or scale formation. It permits designs which warrant mechanization for production line requirements. Time cycles of only a few seconds are maintained by automatic regulation of power for split second heating and quenching which are indispensable for exacting specifications.

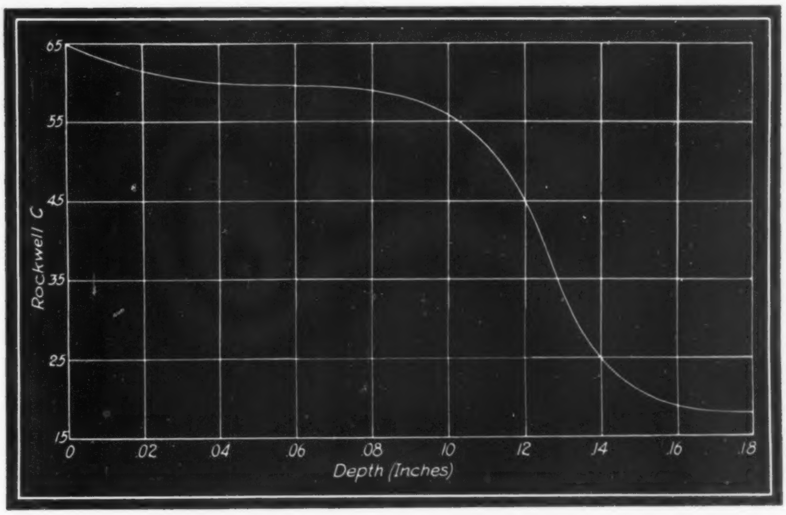
Induction hardening utilizes the phenomena of hysteresis, eddy currents, and skin effect to raise the surface temperature of a piece of steel placed within an energized region to its critical range. Accurate control permits instant quenching to optimum hardness.

Precision obtainable is evidenced by typical inductioning hardening applications to: a crankpin bearing of an SAE 1050 steel, *Fig.* 1, a sprocket with contour controlled to any required specifications, *Fig.* 2, as well as miscellaneous worms, gears, cams, pins, valves, clutch components, etc. shown in *Fig.* 5.

Heating is accomplished by the use of high frequency currents from 2,000 to 10,000 cycles, and in special cases up to 100,000 cycles. Current of this nature in flowing through an inductor produces a high-frequency magnetic field within the region of the inductor. When a magnetic material such as steel is placed within this field, there is a dissipation of energy in the steel which produces heat. The molecules within the steel attempt to align themselves with the polarity of the field and, with this changing thousands of times per second, an enormous amount of internal molec-

ELIMINATION of "bottlenecks" makes desirable this summary of the induction hardening process. Principles and metallurgy are discussed with particular reference to specific applications to machine parts. The obtainable range of properties and the shapes and sizes which lend themselves to being so treated should be of lively interest to the designers of machines

Fig. 3—Curve shows typical hardness distribution throughout the depth of case of induction hardened part



ening Process

uction

W. E. Benninghoff & H. B. Osborn Jr.

Ohio Crankshaft Co.

ular friction is developed, producing heat.

Since another characteristic of high-frequency current is to concentrate on the surface of its conductor, only the surface layers become heated. This tendency, called "skin effect", is a function of the frequency. Other things being equal, higher frequencies are effective at shallower depths. The hysteresis or frictional action producing the heat is dependent upon the magnetic qualities of the steel. An additional source of heat is due to the eddy currents which flow in the steel as a result of the rapidly changing flux in the field. When the temperature has passed the critical point at which the steel becomes non-magnetic, all hysteretic heating ceases and, due to the increase of resistance with temperature, eddy current heating continues at a greatly reduced rate. Because the entire action goes on in the surface layers, only that portion is affected. Original core properties are maintained. Surface hardening is accomplished by quenching when complete carbide solution has been attained in the surface areas. Continued application of power causes an increase in depth of hardness for, as each layer of steel is brought to temperature, the current density shifts to the layer beneath.

Unusual behavior of steel when heated by induction, and the results obtained, merit a discussion of the metallurgy involved. Carbide solution rates of less than a second, higher hardness than that produced by furnace treatment, and a nodular type of martensite are points of consideration that classify the metallurgy of induction hardening as different from thermal methods. Further, surface decarburization and grain growth do not occur because of the short heating cycle.

Induction hardening produces a hardness which is maintained through 80 per cent of its depth and, from there on, a gradual decrease through a transition zone to the original hardness of the steel as found in the unaffected core. A typical curve of hardness versus depth of case is shown in *Fig.* 3. The bond is thus ideal, eliminating any chance of spalling or checking. .

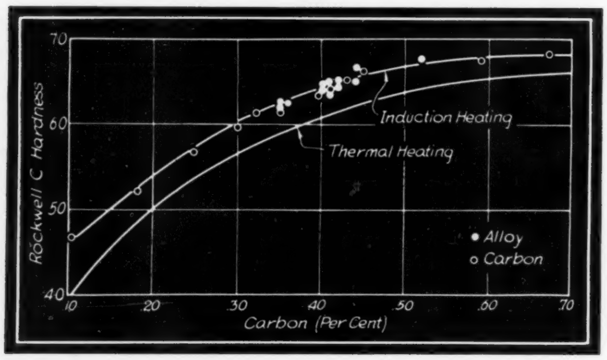
Increased Hardness Obtainable

Typical structure of an induction hardened area has a characteristic appearance. The usual accicular martensitic structure resulting from conventional methods of hardening is definitely absent. Instead, we find a more homogeneous structure; a nodular martensite of obvious greater fineness. Complete carbide solution and homogeneity, as evidenced by maximum hardness and microstructure, can be accomplished with a total heating time of .6 seconds. Of this time, only .2 to .3 second is actually above the lower critical.

To demonstrate that increased hardness is ob-

tainable with induction heating, a number of steels with carbon content ranging from .10 to .70 per cent were tested. Samples of twelve carbon steels (alloy) were used, ranging in size from .5 to 1.5 inch diameter. All were fine grained, (6-8-McQuaid-Ehn) and either normalized or quenched and drawn. Each sample was placed in a commercial inductor block and heated to approximately 50 degrees Fahrenheit above the Ac_3 and instantly quenched. For each sample, 13 to 15, surface readings were taken with a rockwell hardness tester. The average hardness values are plotted in *Fig.* 4, giving the curve which is designated "Induction Heating".

Higher maximum hardnesses result when steel is heated inductively than when heated thermally. This hardness increment is around 2 to 3 points rockwell C at high per cent carbon and 5 to 6 points rockwell C at low per cent carbon. The data indicated by the thermal heating curve are from purely experimental research methods and are not



representative of actual production practice results. The values shown by the induction heating curve are reproducible in an induction hardening unit operating on a production schedule.

The fine nodular and more homogeneous martensite which results from induction hardening is more readily apparent with carbon steels than with alloy steels. This fine structure must have for its origin an austenite which is the result of a more thorough carbide diffusion than is obtained with thermal heating. Practically instantaneous development of critical temperatures throughout the entire microstructure of the alpha iron and iron carbide is particularly conducive to rapid carbide solution and a distribution of constituents which has as its inevitable product a thoroughly homogeneous austenite. Further, the conversion of this structure to martensite will produce a martensite which possesses similar characteristics and a corresponding resistance to wear or penetrating instruments.

Surface hardening equipment for this method of treatment consists of an inductor, quenching auxiliaries, suitable transformers and capacitors, automatic timing controls, and a high frequency generator. In addition, provisions are made for handling parts intermittently or continuously depending upon production requirements. The inductor may be a single turn of copper to fit the piece to be hardened or several turns of copper tubing. Symmetrical inductors may be used to surface harden unsymmetrical objects because of the natural tendency of the high frequency current to follow the contour of the piece. The quenching medium flows through orifices in the inductor as shown schematically in *Fig.* 6. The same timing device that controls the heating cycle operates the quenching cycle to the same degree of accuracy.

Fine Grain Size Is Preferable

Frequency converters of the motor-generator type are used for 2,000, 3,000 and 10,000 cycles, at capacities up to 1,000 kilowatts. Spark-gap oscillators are utilized for higher frequencies up to 100,000 cycles. Higher frequencies are available from vacuum tube oscillators. High frequency current is usually generated at voltages ranging from 200 to 1000. It is then transformed to 20 to 50 volts and fed into the inductor. In some instances, where a multi-turn coil is employed, the transformer is eliminated and connection made directly across the generating source.

It would be impossible at this time to make definite statements concerning all the steels to which the induction hardening method could or could not

Fig. 4 — Left — Consistently higher degrees of hardness in the same material are obtainable by the induction method

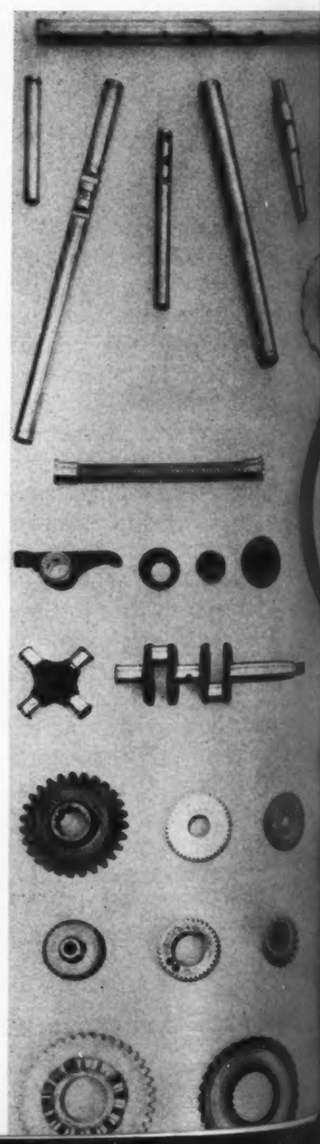


Fig. 5—Right—Typical machine parts to which induction hardening is applicable

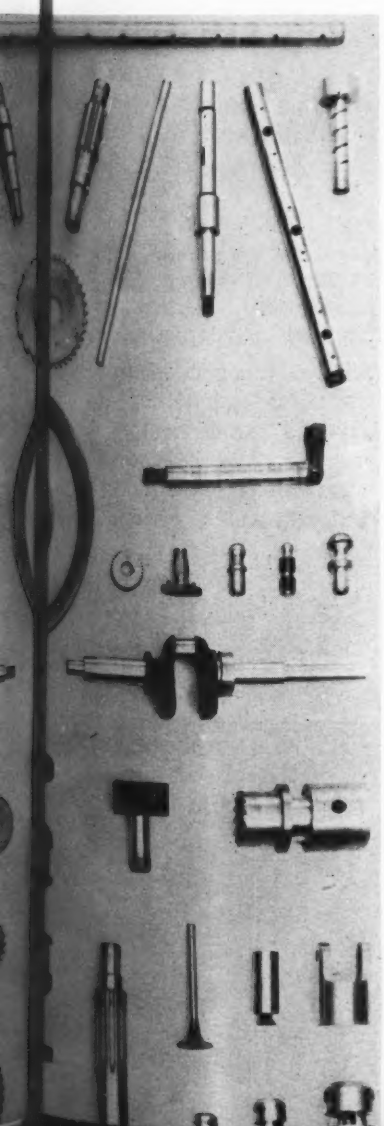
be applied. There are certain requirements, however, which should be kept in mind. The carbon content must be sufficient to produce the desired hardness. A fine grain size is preferable yet not always essential. Due to the excessive demands put on crankshafts, it has been found advisable for such parts to use a heat treated structure which is predominantly sorbitic or a normalized structure having a grain size comparable to the heat treated. Low carbon steels with carburized case, medium and high carbon steels both regular and alloy and ordinary cast iron in a malleable pearlitic condition can all be hardened as desired. Generally speaking, any material which will respond to a heating and cooling operation may be hardened or heat treated by induction.

Contrasted to the small parts which are easily hardened is the problem presented by the hardening of the bearing surfaces on a large diesel crankshaft. Parts such as this are handled in a horizontal position in a tunnel line unit in which each bearing is hardened in an inductor block. Crankshafts which can be handled without the assistance of mechanical equipment are treated in a vertical unit in which three or four bearing surfaces are hardened in one automatically controlled operation.

In the automotive field, economical hardening of camshafts is important. Application of induction hardening, with its selective nature, to the surface hardening of camshafts produces a shaft free from distortion and scale and which is hardened only where necessary. The cams are hardened to 60 rockwell C and the integral helical gear to 52 rockwell C with cast iron, alloy or plain carbon steels.

Procedures involving the assembly of bearing races on automobile axle shafts are expensive. The application of induction hardening to this problem results in surface hardening the shaft at the desired areas and making the shaft act as its own bearing race.

Earlier in this discussion, it was pointed out that induction hardening equipment can be applied readily to production line requirements. Thus some installations utilize continuous rather than intermittent operation. For example, a machine for hardening push rod seats on automobile tappets, illustrates this point. Production can be as high as 7,200 tappets per hour. Further, it is possible to do a progressive job of hardening. Track pins of variable sizes running up to 2 inches in diameter and 10 inches long are surface hardened for their entire length to a selected depth of 3/16-inch with hardness of 64 rockwell C using SAE 1045 steel.



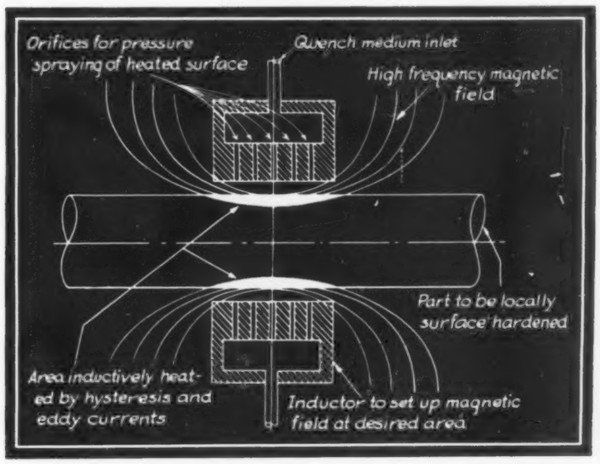
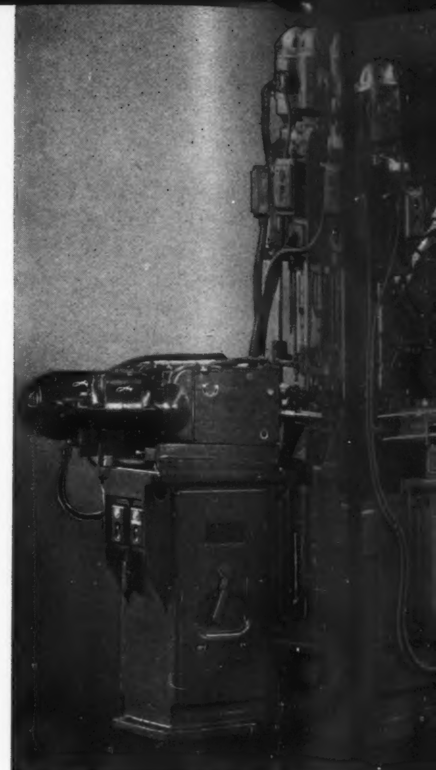


Fig. 6—Right—Sketch illustrates distribution of field and the manner in which the quenching medium is applied

Rocker arm shafts for combustion engines are surface hardened only in areas on which the rocker arms operate. The shaft, in moving through an inductor, is stopped for each area by an indexing mechanism and is then discharged from the machine. A timing device controls the entire operation and makes it automatic once the shaft is placed in the machine.

In addition to the selective surface hardening of steels there have been other interesting and significant applications of induction heating. Hardening a piece of steel and brazing to copper and other metals may be accomplished simultaneously. A small section of a previously hardened object can be drawn or softened to a condition permitting ready machineability. Heating for forging and upsetting has been found to be a particularly satisfactory use for induction heating. The speed with which this may be accomplished has made it readily adaptable to the high production requirements of forming equipment, and scale problems are reduced to a minimum. Tip annealing of brass cartridge shells at the rate of 100,000 per hour is provided with a single induction heating unit.

In conclusion, where speed, accuracy and control are important, induction heating is finding increasing applications. Adoption of induction heating and hardening may provide the solution to a problem, thereby giving better quality and a decrease in overall cost.



Selecting Special Motors

Part VI—Frequent Starting and Reversing Motors

By R. J. Owen and J. J. Kirkish
The Louis Allis Co.

SQUIRREL-CAGE motors readily lend themselves to modification in electrical design to meet conditions of various kinds of service for machine tools and other special machinery. There is perhaps none which calls for more skill in design than required for high rate of reversal. Closely related are applications where the motor must start frequently or plug for a quick stop. The requirements for the motor itself are briefly: It must withstand the mechanical strains which accompany a sudden change in speed or load; the rotor must have a low inertia to minimize the effect in retarding reversal; windings must withstand the high current peaks that occur at each reversal.

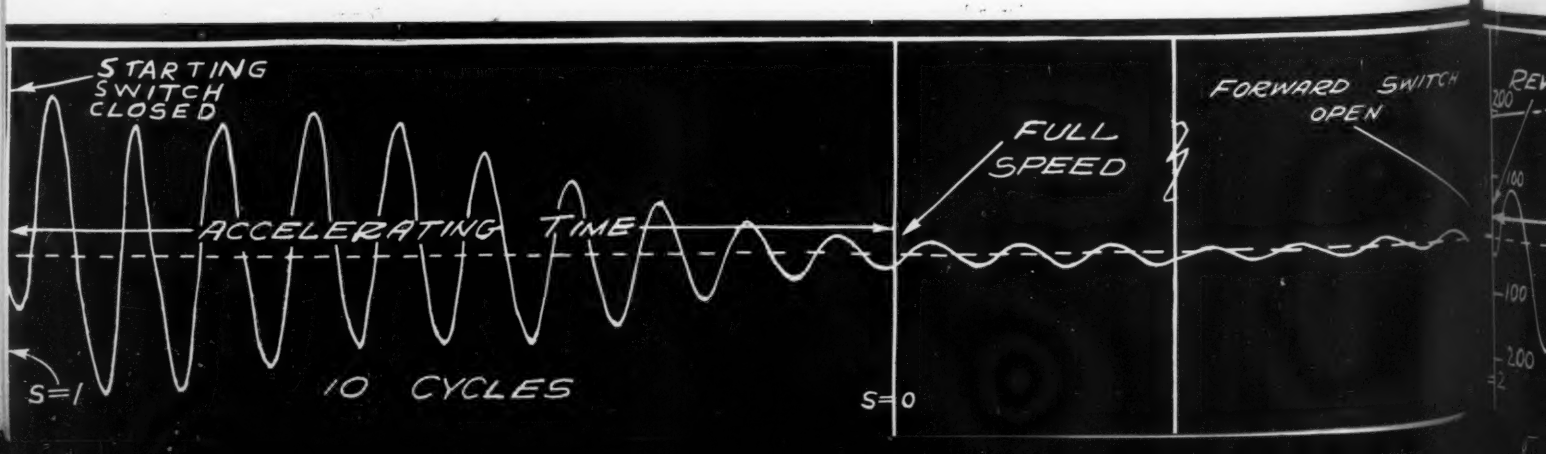
The squirrel cage motor basically meets these requirements and especially so because of the absence of insulated windings in the rotating member, as well as collector rings, commutator and brushes. *Fig.* 1 shows an example of the versatility of this type of motor. Individual motors are used on the spindles as well as the table index drive on this high production automatic machine which may be set up for drilling, tapping, reaming and milling operations. Tapping heads are equipped with flange-mounted motors capable of 12 to 15 reversals per minute. The index drive is handled by a single-speed squirrel cage motor designed for frequent starting and "plug" stopping.

Many factors affect the size and design of motors used for rapid reversing service. They are

1. Inertia of all rotating parts both of rotor and external machine members
2. Speed of the motor and of the load
3. Number of cycles of operation per minute
4. Amount of load torque required during the working cycle of each operation
5. Type of enclosure and methods of ventilation.

From the law of conservation of energy, when an induction motor causes the rotor and its connected load to be accelerated, the I^2R loss in the rotor is exactly equal to the energy stored in the moving parts after acceleration. This well-known fact is the heart of all heating calculations involving frequent starting or reversing. In simplest form it can be expressed as:

$$J_2 = \frac{WR^2 N^2}{4320} \quad \text{.............................} (1)$$



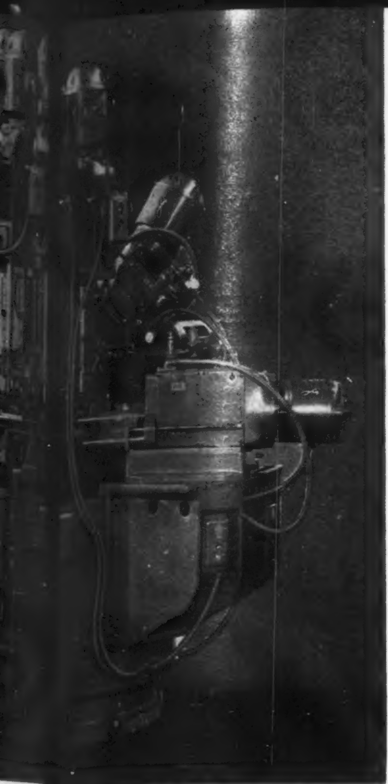


Fig. 1—Left—Individual motors on spindles as well as on the table index drive provide high quantity production on this automatic machine. Spindles are equipped with motors capable of 15 reversals per minute. Index drive is designed for frequent starting and plug stopping

Fig. 2—Right—Effect of WR^2 on permissible number of reversals for self-ventilated, open motor with Class *A* insulation

where J_2 = rotor accelerating loss in joules or watt-seconds, WR^2 = moment of inertia in lb.-ft.^{2} and N = final speed in r. p. m.

It cannot be stressed too strongly that accelerating a mass of a definite and fixed inertia to a specified speed will result in a fixed value of rotor heating. Other losses, however, can be reduced and the heat adequately dissipated. Use of lightweight sheaves, reduction of pulley diameters and other devices can be employed to reduce the inertia of the external system. If most of the WR^2 of the system is in the rotor itself, the design engineer can, as he very often does, aid himself considerably by the use of low inertia rotors. As much material is removed as is consistent with sturdy construction. Sometimes large vent holes are used, sometimes the rotor is built up on special spiders. Still more effective is a reduction in WR^2 by a reduction in rotor diameter. It is easily shown that the following relation exists.

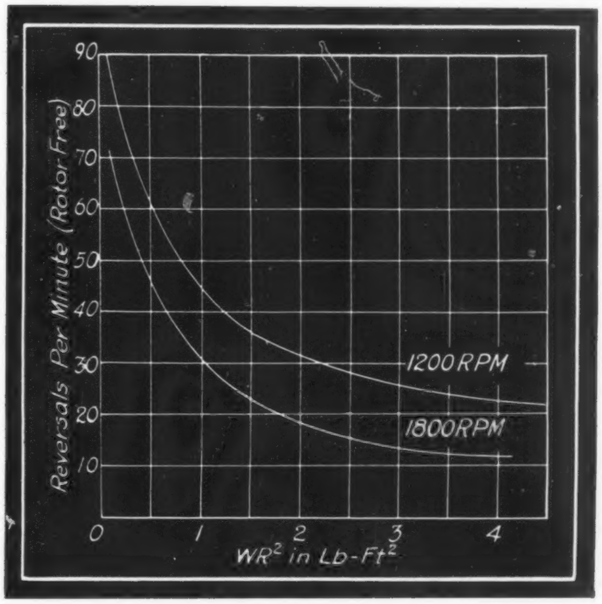
$$WR^2 = K\,D^4 L \quad \text{.............................} (2)$$

where D = diameter of rotor, L = stacking length, and K is a constant of proportionality.

It is possible, within definite limits, to decrease the motor air gap diameter and still maintain an "active layer" by lengthening the core so that DL remains a constant. Thus maintaining equal air gap areas, Equation 2 becomes

$$WR^2 \approx K_1 D^3 \quad \text{..............................} (3)$$

A 20 per cent reduction in rotor diameter results in a 73 per cent reduction in WR^2 with the accompanying decrease in rotor accelerating joules. *Fig.* 2 shows typical effect of WR^2 on the permissible number of reversals for 50 degrees Cent. rise on open motors, self-ventilated, with Class *A* insulation, and rotor free. It must be remembered, however, that the WR^2 in Equation 1 is the moment of inertia of

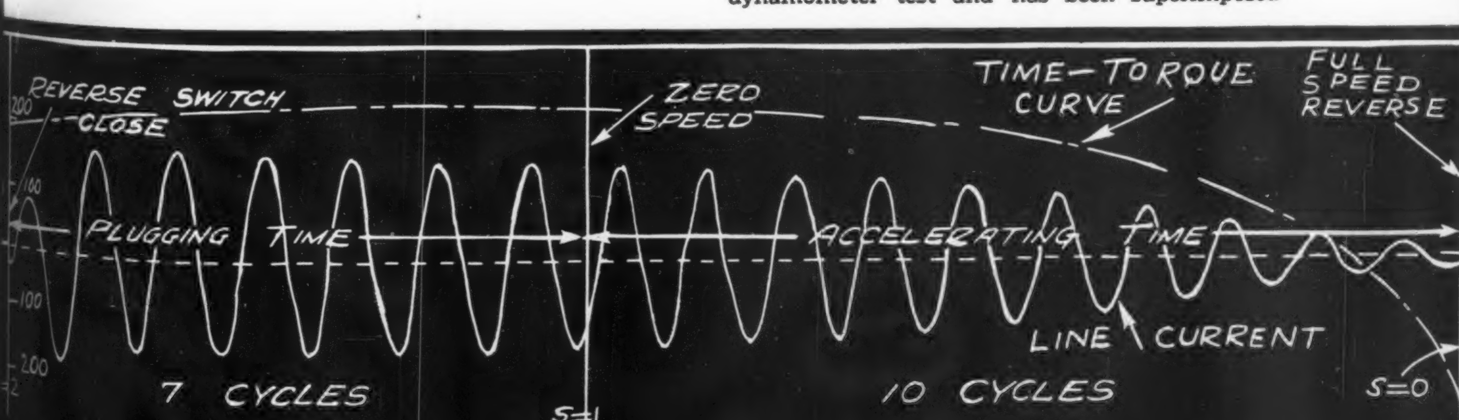


all parts to be accelerated. If the load operates at a different speed than the motor then the WR^2 of the external system must be reduced to that at the rotor speed. For rotating machine members this becomes

$$\text{Equivalent load } WR^2 = WR^2_{(\text{load})} \times \left(\frac{RPM_{(\text{load})}}{RPM_{(\text{motor})}} \right)^2 \quad \text{..........} (4)$$

When the motion is one of translation then

Fig. 3—Below—Oscillogram showing accelerating, plugging and reversing currents on a 20-horsepower motor. Time-torque curve has been calculated from dynamometer test and has been superimposed



$$\text{Equivalent load } WR^2 = 91.2 \frac{WV^2}{N} \text{ .... (5)}$$

where W = weight of moving mass in lb., V = velocity of the member in ft. per second, and N = motor r.p.m.

Equations 1 and 4 show how important is the question of motor and spindle speeds. Thus pulley ratios should be kept as low as possible. Similarly, the lower the motor speed the greater will be the number of permissible reversals. There are partially counteracting effects, however, such as an increase of other losses occasioned by high excitation and correspondingly poorer ventilation.

TABLE I illustrates several of the points previously mentioned. It shows the number of reversals

TABLE I

Permissible Reversals per Minute for Various Speeds and High External Inertia

Motor Insulation	Motor Speed	SPINDLE SPEEDS 6000/3000	3300/1650	1850/925	980/490
Class A......	1800	4	7	10	11
Class A......	900	7	9	17	19
Class B......	1800	5	9	12	14
Class B......	900	8	15	20	23

that can be obtained from a multispeed motor having 1-winding, 2-speed, constant-torque 1800/900-r.p.m., 1.5/.75-h.p., built in the 225 N.E.M.A. frame. Spindle speeds range from 6000 to 490 r.p.m. The permissible number of reversals on both speeds are shown for several spindle speeds with large external WR^2 using cast iron sheaves. Inertia of the system includes: Rotor WR^2 = 70 lb.-ft.^{2}; motor pulley .58 lb.-ft.^{2}; countershaft pulley and parts at 6000/3000 r.p.m. 2.60 lb.-ft.^{2} at 3300/1650 r.p.m. .79 lb.-ft.^{2}, at 1850/925 r.p.m. .25 lb.-ft.^{2} and at 980/490 r.p.m. .07 lb.-ft.^{2}

If, however, cast aluminum sheaves are substituted for cast iron there is a considerable increase in the allowable reversals due to the reduction of the external WR^2. This is shown in TABLE II.

TABLE II

Permissible Reversals per Minute for Various Speeds and Low External Inertia

Motor Insulation	Motor Speed	SPINDLE SPEEDS 6000/3000	3300/1650	1850/925	980/490
Class A......	1800	7	11	13	14
Class A......	900	12	20	23	25
Class B......	1800	9	14	17	18
Class B......	900	15	24	29	31

An illustration of the extent to which inertia can be reduced is shown in *Fig.* 5. Drive for the precision lathe is a four-speed, built-in motor designed for frequent stopping and plugging. The rotor is mounted direct on the spindle with cooling from a separate blower unit. In addition to cooling, even though the drive motor is stopped, the inertia of the fan is not a part of the drive and therefore does not require energy for stopping and starting. This feature alone permits an increase in the number of starts of between 50 and 100 per cent.

For any dissipation of energy in the rotor during acceleration there will result a corresponding copper loss in the stator. By transformer action any value of rotor current I_2 (when expressed in equivalent stator terms) is balanced by a stator current I' exactly equal and opposite to it. There is, however, an additional component of current, required for excitation, which added vectorially to I' gives the total value of stator current I_1. It is difficult to express the ratio I_1/I_2 in any simple terms as it varies with the motor slip and is larger with a

TABLE III

Values of Energy Dissipation Factor

Number of Poles	Syn. Speed 60 Cycles	Factor C_1
2	3600	1.00
4	1800	1.05
6	1200	1.10
8	900	1.20
12	600	1.30

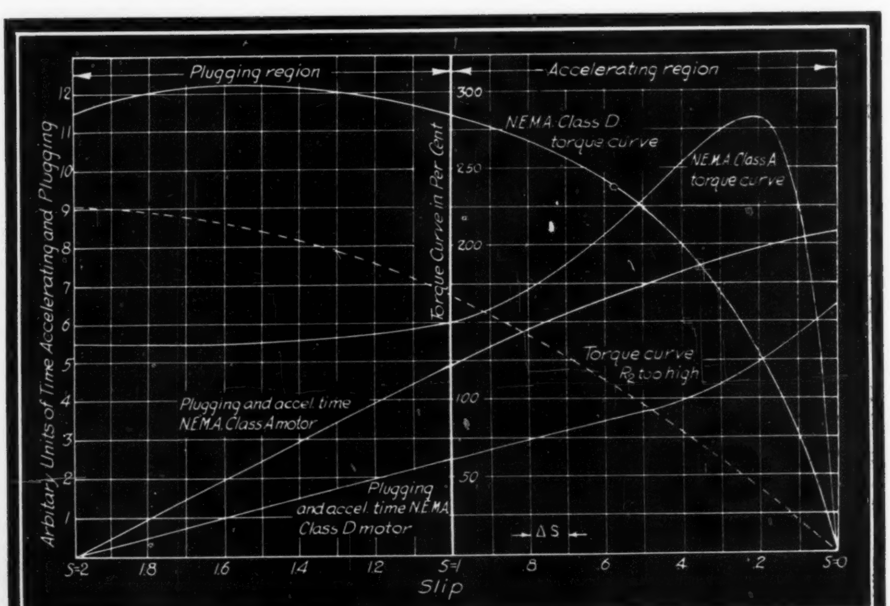


Fig. 4—Torque curves for N.E.M.A. Class A and Class D squirrel-cage motors

greater number of poles. Thus for any rotor copper loss $I_2^2 r_2$ there results a corresponding stator copper loss $C_1 I_2^2 r_1$ (where C_1 represents the square of the I_1/I_2 ratio). For simplicity, C_1 can be taken from TABLE III. Hence for the accelerating period

$$J_1 = \frac{WR^2 \times N_s^2}{4320} \left(C_1 \frac{r_1}{r_2} \right) \qquad (6)$$

where J_1 = stator accelerating loss in joules, r_1 = stator resistance, and r_2 = rotor resistance.

Stated in its more general form: The total copper loss in both stator and rotor when the motor speed is changed from initial slip S_i to final slip S_f is

$$J_1 + J_2 = \frac{WR^2 N_s^2}{4320} \left(1 + C_1 \frac{r_1}{r_2} \right) (S_i^2 - S_f^2) \text{ joules} \qquad (7)$$

On machine tool applications requiring reversals of from five to fifty per minute the method of stopping will determine, to a large extent, the permissible cycles of operation. By far the most common method is by plugging because its simplicity and ease of operation make it preferable to mechanical or direct-current braking. Unfortunately from the heating standpoint it is the worst of the three. Mechanical braking does not, of course, contribute any loss to the motor. Dynamic braking, even with sufficient direct-current to stop the motor in the same interval of time as with alternating-current plugging, causes less heating.

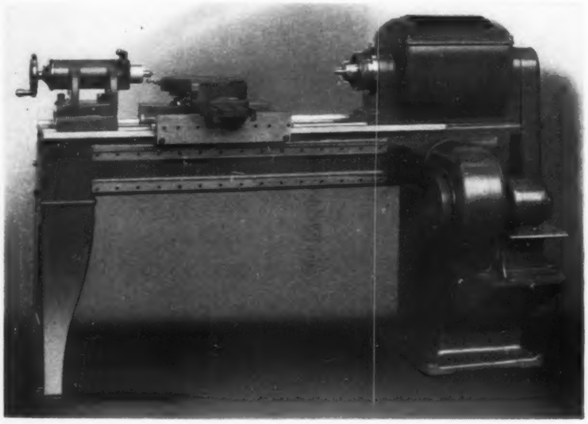


Fig. 5—High speed precision lathe driven by a four-speed built-in motor designed for frequent starting and plug stopping. Rotor is mounted direct on spindle and cooling is provided by separate blower to reduce inertia of system. Auxiliary ventilation increases rate of operation from 50 to 100 per cent

At the instant of plugging an induction motor, the slip equals 2. By referring to Equation 7 it is readily seen that for plugging to standstill ($S_i = 2$, $S_f = 1$) the decelerating loss in both primary and secondary are

$$J_1 + J_2 = 3 \times \frac{WR^2 N_s^2}{4320} \left(1 + C_1 \frac{r_1}{r_2} \right) \text{ joules} \qquad (8)$$

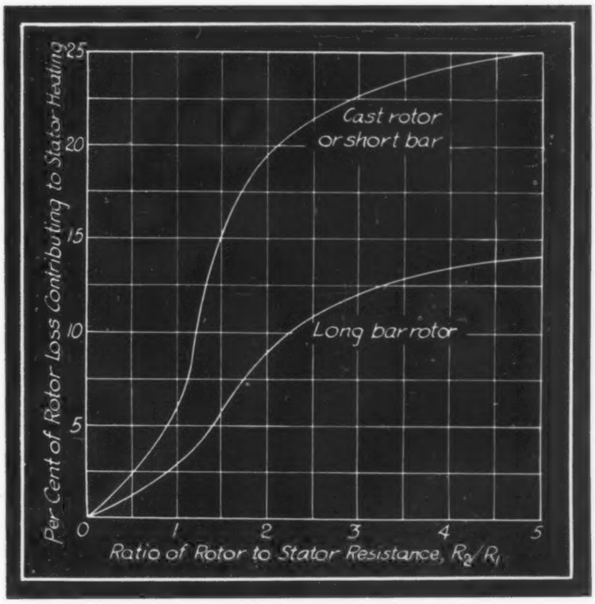


Fig. 6—Rotor loss contributing to heating for short and long bar rotors

For one complete reversal, i.e., from standstill to full speed and plug back to standstill,

$$J_1 + J_2 = \frac{WR^2 N_s^2}{1080} \left(1 + C_1 \frac{r_1}{r_2} \right) \text{joules} \qquad (9)$$

In conjunction with Equation 7 the special cases of S_i — S_f for single-speed motors are shown in TABLE IV. The last item in the table applies only to rotor decelerating joules as the stator loss is determined by the direct current.

TABLE IV

Slip Factor for Single-Speed Motors

Condition	$S_i^2 - S_f^2$
From standstill to synchronous	1
Plug from full speed to standstill	3
Complete reversing cycle	4
Braking with DC in stator	1

There still remains to be mentioned the full-speed load losses which although small in comparison with those of acceleration and plugging should still be considered. These are the usual induction motor losses consisting of iron loss, stator and rotor copper loss and stray.

Summing up the total energy dissipated during one reversing cycle

$$J_{(total)} = \frac{WR^2 N_s^2}{1080} \left(1 + C_1 \frac{r_1}{r_2} \right) + J_{FE} + J_{(load)} \text{ joules} \qquad (10)$$

where J_{FE} = iron loss joules = watts iron loss $\times$ time of one cycle, and $J_{(load)}$ = load loss joules = stator and rotor load loss $\times$ full speed running time.

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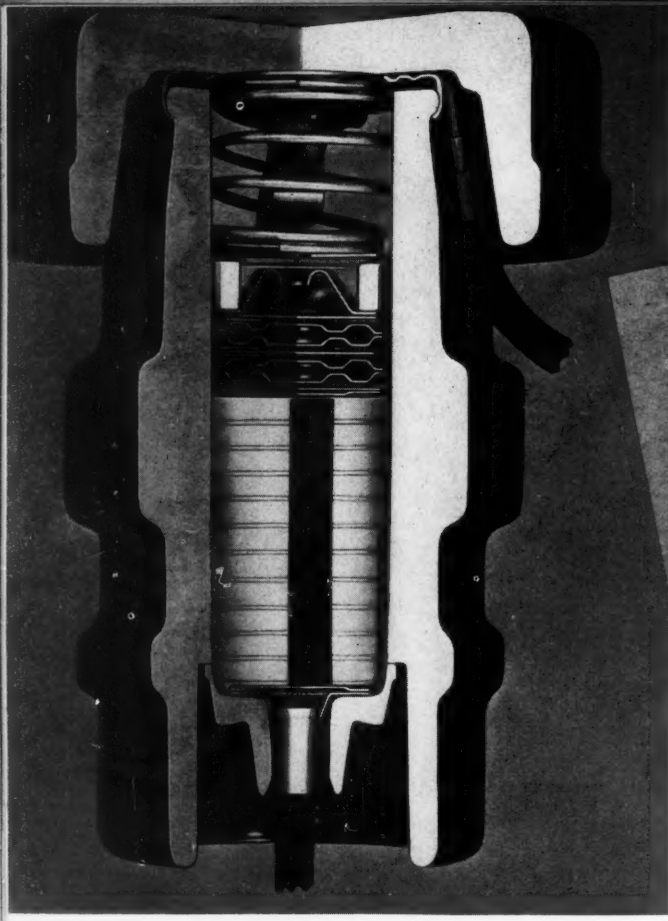


Fig. 1—Statically loaded helical spring used in lightning arrester to maintain spacing

Calculating Sprin[gs] for static loadin[g]

By A. M. Wahl
Westinghouse Research Laboratories

ALLOWABLE loads on springs under static or infrequently repeated loading may be determined readily by a method of analysis based on the assumption that stress concentration effects due to bar curvature may be neglected. This method of choosing working stress and allowable loads as well as some of the factors which a designer should consider in selecting springs for static load applications is discussed in this article. The load required to produce complete yielding of the spring is also estimated as a function of yield stress and spring index.

Statically loaded helical springs may be defined as those subject to constant loads or loads repeated but a relatively few times during the life of the machines or structures of which the springs form a part. This type of loading is in contrast to the case of fatigue loading where the load may be repeated millions of times during the life of the spring, as in the case of valve springs used in internal combustion engines.

Fatigue Loading Not Considered

Examples of springs under static loading are: springs used to provide gasket pressure in certain applications such as condenser bushing caps, springs used in circuit breaker mechanisms which operate infrequently, safety valve springs in applications where the valve opens but seldom, and many other similar applications. An important application of a statically loaded spring is a lightning arrester illustrated in *Fig.* 1. Here the function of the spring is to maintain a definite space relationship between the disks of the arrester regardless of temperature change.

When a spring is subject to fatigue or repeated loading, failure may occur by the development of a fatigue crack which causes eventual fracture of the spring. If however, the spring is subject to a static load, this type of failure will not occur. In this case, however, the designer must guard against excessive loss in load which occurs when too high working stresses are used. If a small amount of load loss can be tolerated, a higher working stress may be used than would be the case otherwise.

Small Index Allows Higher Stresses

A factor which is of particular importance in the design of springs subject to static loading is the *spring index,* i.e., the ratio between coil diameter and wire diameter. Where the spring index is small the highest stress is concentrated near the inside of the coil. When the load is calculated by taking this stress into account, as will be seen later, a higher value is permissible in such cases than would be the case for springs of larger indexes.

In considering a spring subject to a static load and assuming the peak stress is below the elastic limit, the stress distribution along a transverse diameter for a spring of small index is shown approximately by the line *bc* in *Fig. 2a.* The peak

shearing stress ab in this case is calculated by the commonly used formula[1]

$$S_{s\,(\max)} = \frac{16Pr}{\pi d^3} K \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

$$K = \frac{4c-1}{4c-4} + \frac{.615}{c} \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where P = load on spring, r = coil radius, d = bar or wire diameter, and $c = 2r/d$ = spring index.

From *Fig.* 2*a* it is seen that for a spring of small index the stress ab on the inside of the coil is much larger than the stress $a'c$ on the outside of the coil, i.e., most of the high stress is concentrated near point a. This means that a condition of stress concentration exists, as is shown graphically by *Fig.* 3*a* where the peak stress is concentrated in the relatively small shaded area near a.

When the spring index is large, conditions are considerably different as shown in *Fig.* 2*b*. Here the stress ab on the inside of the coil is only a little larger than the stress $a'c$ on the outside. In this case the peak stress is given approximately by taking $K = 1$ in Equation 1. This follows from Equation 2 which reduces to $K = 1$ for large values of c.

Larger Index Springs More Uniformly Stressed

In the case where the index is large the highest stresses are located in the ring-shaped shaded area shown in *Fig.* 3*b* instead of being concentrated in a relatively small region near the inside of the coil as is the case where the index is small, *Fig.* 3*a*. In other words, in the case of the spring of small index only a relatively small portion of its cross-sectional area is subjected to stresses near the peak, while in the case of the spring of large index a relatively large part of the cross section is subject to such stresses.

Hence, if the load is increased so that yielding occurs over the entire cross-section of the bar or wire, it is clear that the spring of small index will be able to carry a much larger load than would be expected on the basis of the maximum stress calculated from Equation 1 which assumes purely elastic conditions. The reason for this is that after the elastic limit is passed and yielding begins most of the cross-section will be effective in carrying load even for small indexes. Since a good share of the cross-section of the spring of large index is already subject to stresses near the peak, the increase in load necessary to produce complete yielding over the entire section will not be so great as in the case of the small-index spring, where only a small part of the section is initially subject to stresses near the peak.

Usually Neglect Stress Concentration Effects

Most spring materials have considerable ductility (although, of course, much less than have structural materials). For example, a tension stress-strain diagram of a typical chrome-vanadium steel as used for small springs[2] is shown in *Fig.* 4. It is seen that this has a shape of tensile stress-strain diagram characteristic of a ductile material with a fairly sharply defined yield point. Since most of the useful spring materials have elongations greater than 5 per cent in 2 inches, and stress-strain diagrams similar to that of *Fig.* 4, (although they may have a greater slope after the yield point has been passed) it appears reasonable to treat them a ductile materials. In such cases, where static loads are involved, it is usual practice to neglect stress concentration effects in design[3]. This means that the additional stress due to the bar curvature which is present particularly for springs of smaller index may be neglected in calculating the stress under static load conditions.

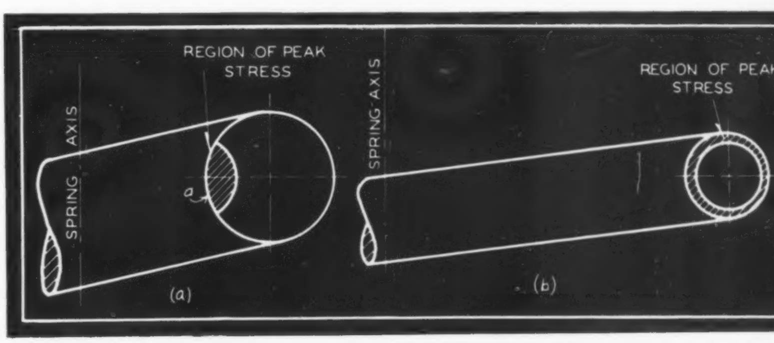


Fig. 3—Relative distribution of regions of peak stress in cross section of a bar of helical spring. At *a* and *b* are shown small and large index springs respectively. For the spring of small index most of the stress is concentrated near the inside of the coil

To calculate the stress in a spring by neglecting stress concentration due to bar or wire curvature, it is possible to proceed as follows: Assuming a helical compression or tension spring under a load P and neglecting effects due to the end turns, the

[1] See MACHINE DESIGN, February, 1938, for a further discussion of this formula.

[2] See "Permissible Stress Range for Small Helical Springs" by F. P. Zimmerli, *Engineering Research Bulletin No.* 26, University of Michigan, for other similar diagrams.

[3] See paper by Soderberg on "Working Stresses", *Transactions A.S.M.E.*, 1933, A-P-M 55-16.

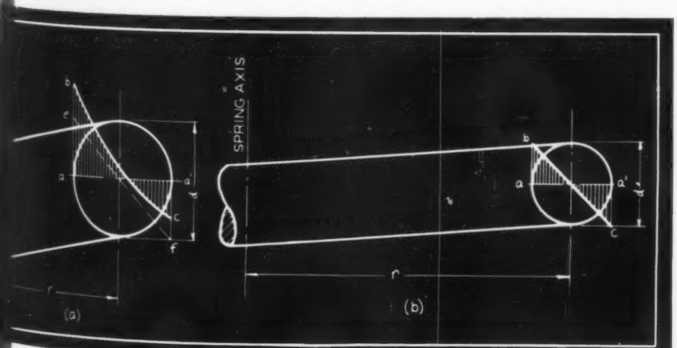


Fig. 2—Distribution of stress along transverse diameter of bar of helical spring assuming elastic conditions. At *a* is shown a small index and at *b* a large index

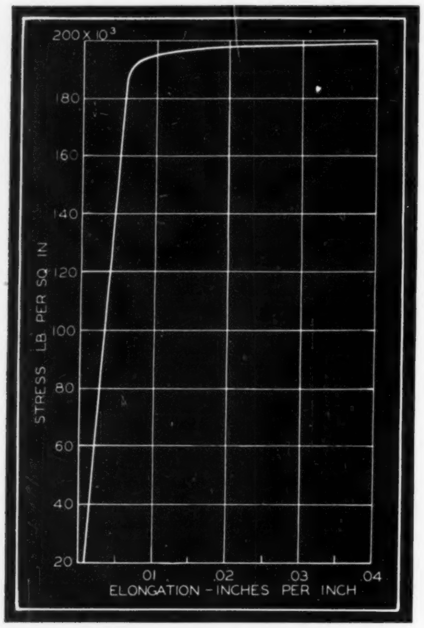


Fig. 4 — Stress - strain diagram for chrome vanadium spring steel

torsion moment at any point along the bar will be equal to Pr while the direct shear will be equal to P. The torsion stress-distribution along a transverse diameter due to the moment Pr will be as shown in *Fig.* 5*a* while the peak stress due to this moment will be that given by the well-known formula:

$$S_{s_1} = \frac{16Pr}{\pi d^3} \qquad (3)$$

On this stress must be superimposed the shear stress due to the direct shear load P, which may be considered as uniformly distributed over the cross-section[4]. This stress will be assumed distributed as shown in *Fig.* 5*b* and is

$$S_{s_2} = \frac{4P}{\pi d^2} \qquad (4)$$

The maximum stress will be obtained by superposition of the distributions of *Fig.* 5*a* and *b* giving a resultant distribution shown in *Fig.* 5*c*. The maximum stress, thus obtained by neglecting stress concentration effects, is

$$S'_{s\,(max)} = S_{s_1} + S_{s_2} = \frac{16Pr}{\pi d^3} + \frac{4P}{\pi d^2} \qquad (5)$$

This equation may be written

$$S'_{s\,(max)} = \frac{16Pr}{\pi d^3} K_s \qquad (6)$$

where

$$K_s = 1 + \frac{.5}{c} \qquad (7)$$

For convenience in calculation K_s is plotted as a function of the spring index c as shown in the curve of *Fig.* 6.

[4] Actually there will be some nonuniformity in the distribution of the shear stress but since this will have a similar effect to that of stress concentration due to bar curvature it will be neglected.

Thus for static loads it is suggested that Equation 6, derived on the basis that stress concentration may be neglected, should be used. To form an idea of the margin of safety of the spring against yielding, the stress computed by Equation 6 should be compared with the yield point of the material in torsion, which for most spring materials may be taken to be about 75 per cent of the tension yield point.

Calculation of Load for Complete Yielding

There is a somewhat different approach to the problem of designing a spring for static load conditions. This method is based on the consideration of the load required to produce complete yielding of the material in the spring. If it is assumed that a spring material gives a stress-strain curve similar to that of *Fig.* 4, it may be expected that after exceeding the yield-point the distribution of stress across a transverse diameter will be something like that shown in *Fig.* 7*a* for a spring of small index and in *Fig.* 7*b* for one of large index. Actually it may be expected that for many materials some rise in the stress-strain curve after passing the yield point will take place due to cold working effect so that actually the curves of *Fig.* 7 will be approximately trapezoidal in form. For this purpose, however, the assumption of a rectangular distribution (which lends itself to simplicity in analysis) will be sufficiently exact. Because of the necessity for carrying a considerable direct shear load particularly where the spring index is small, it may be expected that in such cases the area A_1 will be greater than A_2, *Fig.* 7*a*. On the other hand, if the index is large these two areas will be about the same, *Fig.* 7*b*. Due to this effect, the point O' where the stress is zero is shifted by an amount ϵ_0 from the geometrical center O of the bar or wire as shown in the diagram.

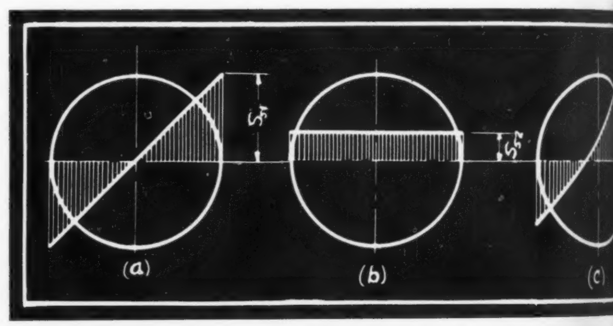


Fig. 5—Superposition of stresses in a helical spring Stress concentration due to bar curvature is neglected. At *a*, *b* and *c* are shown stress due to torsion moment, stress due to direct shear, and superposition of stresses due to direct shear plus torsion

Under our assumption that the spring material

is perfectly plastic, i.e., it yields at a constant stress S_{sy} equal to the torsional yield point, it appears reasonable to take a distribution as shown along a diameter AA' (*Fig.* 8*a*) which is inclined at an angle θ to a diameter BB transverse to the spring axis. In this picture the point of zero stress is shifted from the geometrical center O to a point O' at a distance ϵ from O. As a first approximation ϵ will be taken equal to $\epsilon_o \cos\theta$. This condition satisfies the end conditions as follows: for $\theta = 0$, $\epsilon = \epsilon_o$ (*Fig.* 7*a*) and $\theta = 90$ degree, $\epsilon = 0$, which must be true from symmetry conditions.

Calculations Are Within 2½ Per Cent

Considering a small element subtended by two radial lines aa' and bb' making a small angle $d\theta$ with each other (*Fig.* 8*b*), a small element E at a distance ρ from the center O will have an area $\rho d\rho d\theta$ The small moment dM about O produced by the shear stresses acting over the area enclosed by the radial lines aa' and bb', as indicated graphically in *Fig.* 8*b*, will be

$$dM = 2\, d\theta \int_{\epsilon_o \cos\theta}^{d/2} S_{sy}\rho^2 d\rho \quad \text{.....................(8)}$$

This is obtained by multiplying the area of the element E by the shear yield stress S_{sy} and by the lever arm ρ of the resultant load about O. Integration is between $\rho = \epsilon_o \cos\theta$ and $\rho = d/2$, the outer radius of the bar, since the forces between $\rho = 0$ and $\rho = \epsilon_o \cos\theta$ do not produce a resultant moment. Thus integrating the expression shown in Equation 8 we find

$$dM = \frac{2}{3} S_{sy} \left(\frac{d^3}{8} - \epsilon_o^3 \cos^3\theta \right) d\theta \quad \text{.........(9)}$$

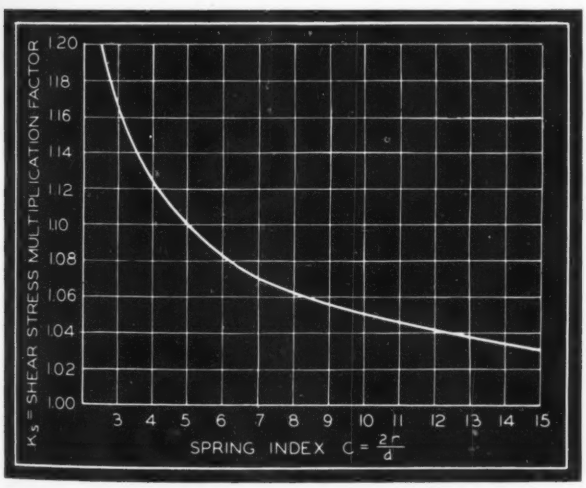


Fig. 6—Curve for finding shear stress multiplication factor. This factor takes into account effects due to direct shear load but not to bar curvature

The total moment will be

$$M = 2\int_o^{\pi/2} dM$$

$$= \frac{4}{3} S_{sy} \int_o^{\pi/2} \left(\frac{d^3}{8} - \epsilon_o^3 \cos^3\theta \right) d\theta \quad \text{.....(10)}$$

Integrating and taking the moment $M = P_y r$, where P_y is the load required to produce complete yielding,

$$P_y = \frac{\pi d^3 S_{sy}}{12r}\left(1 - \frac{32\epsilon_o^3}{3\pi d^3} \right) \quad \text{.................(11)}$$

or, solving for S_{sy}

$$S_{sy} = \frac{3}{4}\left(\frac{16P_y r}{\pi d^3} \right)\left(\frac{1}{1 - \frac{3.4\epsilon_o^3}{d^3}} \right) \quad \text{........(12)}$$

or

$$S_{sy} = \frac{3}{4}\left(\frac{16P_y r}{\pi d^3} \right) K'_s \quad \text{...................(13)}$$

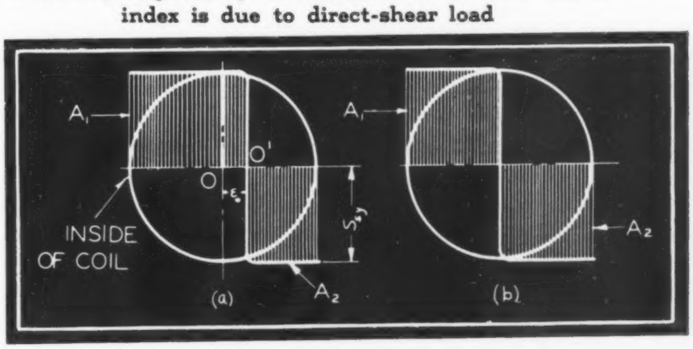
where

$$K'_s = \frac{1}{1 - \frac{3.4\epsilon_o^3}{d^3}} \quad \text{........................(14)}$$

Besides this, a second equation is needed to determine the two unknowns, P_y and ϵ_o. This relation is obtained from the condition that the summation of the components in the vertical direction of the resultant loads on each of the elementary strips in *Fig.* 8*b* must equal the shear force P_y which acts on the spring.

Consider a single strip formed by the angle between two adjacent diameters aa' and bb', *Fig.* 8*b* The resultant load on this strip will be twice the area of the strip between the point O and the point where $\rho = \epsilon_o \cos\theta$ since the remaining loads will balance out. The vertical component will be this

Fig. 7—Assumed distribution of stress under plastic conditions for springs of different indexes. Small index and large index spring sections are shown at *a* and *b*, respectively. Difference in area for small index is due to direct-shear load



value multiplied by $cos\theta$. Thus

$$dQ = 2cos\theta d\theta \int_0^{\epsilon_0 cos\theta} S_{sy}\rho d\rho$$

or integrating between the given limits

$$dQ = S_{sy}\epsilon_0^2 cos^3\theta d\theta$$

The total load P_y necessary to produce complete yielding will then be twice the integral of these elementary forces dQ taken between 0 and $\pi/2$

$$P_y = 2 \int_0^{\pi/2} S_{sy}\epsilon_0^2 cos^3\theta d\theta$$

or integrating and substituting limits

$$P_y = \frac{4}{3} S_{sy}\epsilon_0^2 \qquad (15)$$

By substituting the value of P_y given by Equation 15 in Equation 12, the following relation exists between the spring index c and the ratio ϵ_0/d

$$\frac{\epsilon_0^3}{d^3} + .748 \frac{c\epsilon_0^2}{d^2} - .294 = 0 \qquad (16)$$

This equation yields values of ϵ_0/d as a function of the spring index c. By using these values in Equation 14, the factors K_s' may be determined as a function of c. The final results show that K_s' differs from K_s as given by Equation 7 or *Fig.* 6 by not more than about 2½ per cent for spring indexes greater than 3, values of K_s' being slightly below corresponding values of K_s. Hence the curve of *Fig.* 6 or Equation 7 may be used with sufficient accuracy. Equation 14 may then be written

$$S_{sy} = \frac{3}{4}\left(\frac{16P_y r}{\pi d^3}\right) K_s \qquad (17)$$

or solving for P_y

$$P_y = \frac{4}{3}\left(\frac{\pi d^3 S_{sy}}{16 r K_s}\right) \qquad (18)$$

Equation 6 which holds for elastic conditions may also be solved for P as follows:

$$P = \frac{\pi d^3 S'_{s\,(max)}}{16 r K_s} \qquad (19)$$

If $S'_{s(max)}$ is equal to the yield stress in torsion S_{sy}, comparison of Equations 18 and 19 shows that, under the assumption made, the load required to produce complete yielding of the spring material will be about 4/3 the value of the load at which yielding starts, calculated by Equation 19 which neglects stress concentration effects due to bar curvature. This means that pronounced yielding of the spring may be expected at a load about 33 per cent higher than that at which yielding first begins.

It should be remembered that Equation 18 was derived under the assumption of perfect plasticity which is not rigorously fulfilled since the stress-strain diagrams for most spring materials show some tendency to rise after the yield point is passed. Nevertheless, the results obtained by this method of calculation do give an indication of what may be expected after the yield point has been passed.

Practical Applications of Formulas

It is common practice at present to base the design of springs on Equation 1 which yields a value of stress assuming elastic conditions and taking into account the stress concentration effects due to bar or wire curvature and the stress increase due to direct shear. Hence the use of this formula is reasonable where springs are under fatigue or repeated loading. In any case results obtained by this formula should be on the safe side; however,

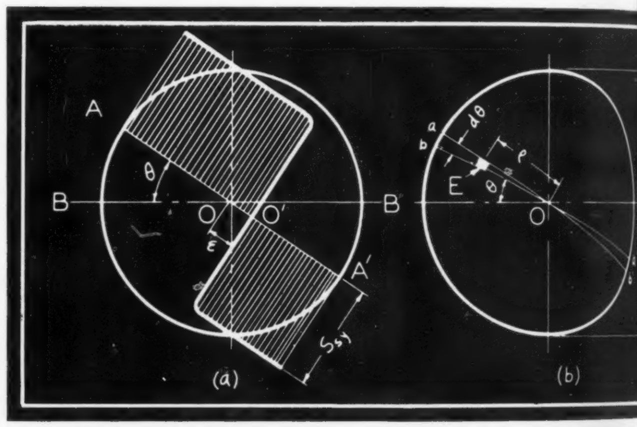


Fig. 8—Distribution of stress along diameter of a bar section at angle Θ

for applications where the load is static so that fatigue does not enter and where a slight amount of yielding may be tolerated the stress may be figured from Equation 6 where stress concentration is neglected using the factor K_s instead of K.

If the load is taken from a spring table or chart based on Equation 1, the stress on which the table is based may simply be divided by a factor $K_c = K/K_s$ to obtain a stress which may be compared with the torsion yield point or elastic limit. Values of K_c (which may be considered as a stress concentration factor, taking into account effects due principally to bar curvature) are given in *Fig.* 9 as a function of spring index. Comparison of the calculated stress with the yield stress in torsion shows the relative margin of safety for static loading.

As an example of design procedure based on this method: A spring has an index of 3 and is chrome vanadium steel having a tension yield stress of around 190,000 pounds per square inch and a yield stress in torsion about 57 per cent of this or 110,000 pounds per square inch. Assuming a factor of safety

of 1.5 based on the yield stress in torsion is to be used, the working stress for the static load condition from Equation 6 would be 110,000/1.5 = 73,000 pounds per square inch. For an index of 3, the factor K_c = 1.35 (*Fig.* 9); hence the allowable stress using Equation 1 would be 73,000 (1.35) = 98,000 pounds per square inch. If charts based on this equation are used (such as those given in MACHINE DESIGN, February 1938) the other spring proportions such as active turns, coil and wire diameters, free and solid heights, may easily be determined.

Is Rational Basis for Design

As a second example, a spring has an index of 15 with other conditions the same as in the previous example. For an index of 15, the factor K_c = 1.06 from *Fig.* 9. Again assuming an allowable stress figured by neglecting stress concentration, equal to 2/3 the yield stress in torsion or 73,000 pounds per square inch, the allowable stress from Equation 1 would be 73,000 (1.06) = 77,400 pounds per square inch. This stress is considerably lower than the allowable value of 98,000 pounds per square inch for the spring of index 3 considered in the previous example. This illustration shows how the working stress would vary with the spring index, assuming that the same margin is being maintained between the working load and the load required to cause complete yielding of the spring.

Thus, in applications where springs are subject to a constant load or to a load repeated a relatively

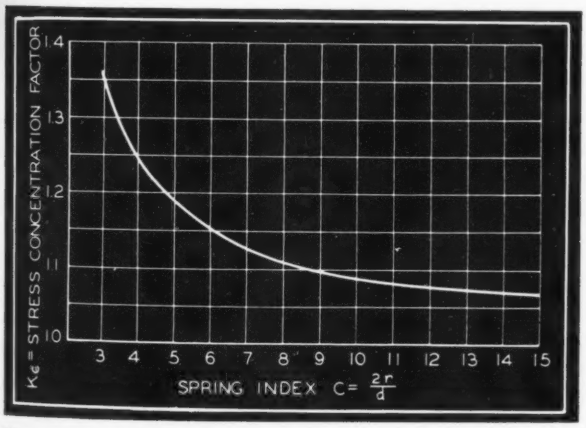


Fig. 9—Curve for finding stress-concentration factor

few times, the method of analysis discussed appears to yield a rational basis for design. If the working stress is based on the maximum stress in the spring —calculated by assuming elastic conditions and taking stress concentration effects into account—a higher stress may safely be used for the springs of smaller index.

Although effects due to creep or relaxation at elevated temperatures are not considered here, it may be expected also that similar conclusions would hold in this case, i.e., the allowable stresses calculated by assuming elastic conditions would be higher for the springs of smaller index.

Light Tanks Are Designed To Take It

LIGHT tanks are now rolling off a regular production line at the American Car and Foundry Co. plant. To meet a regular production schedule it was necessary for the company to process its own case-carburized armor plate. About 75 per cent of the total weight of a tank is made up of this heavy plate. Over 2000 rivets are utilized in each tank and,

being air-hardening steel, they introduced considerable manufacturing problems, especially when driven with a pneumatic hammer.

To design a vehicle whose ultimate employment is the roughest possible usage requires careful design and great precision in manufacture. In fact drilled holes do not suffice; reamed ones are required. A tank is expected to traverse rutted fields and tangled woods, to splash across unbridged streams, to be subjected at times to having its full weight thrown upon itself. Therefore it must be sturdily constructed but of minimum weight for mobility and for maximum utilization of the engine horsepower.

Minimum weight to accomplish the purpose intended is an essential feature in the design of tanks. For this reason all parts are as light in weight as possible and high alloy steels, heat treated to develop full strength are utilized.

There is no chassis, the armor plate furnishing the structural frame as well as protection from enemy fire. This requires not only close-fitted parts but also flat plate. No shims are allowed in the engine supports or in other mechanisms attached to the plate.

Exclusive of the engines and accessories, 14,318 individual pieces make up one tank. Principal assemblies include transmission, hull, turret, suspensions, caterpillar track, clutch, gun and mounts.

Applying Theory of Elasticity

Part V—Curved Beams

By R. E. Orton
Chief Engineer
Acme Steel Co., Chicago

CURVED beams subjected to uniform bending moments are covered in this section as the remaining case of ring distribution. The preceding section discussed ring distribution of stress produced by loads applied normally to the boundaries of a full ring. This present article will conclude with the general loading of curved beams.

In *Fig.* 36 is illustrated a curved beam under a uniform bending moment M. Mathematically the

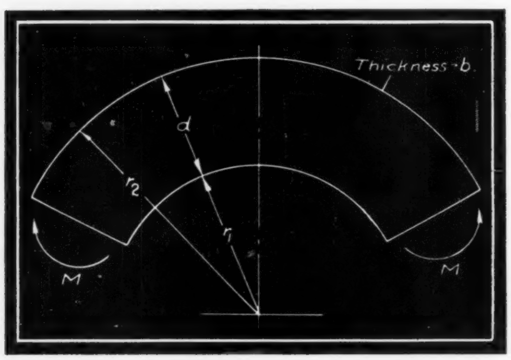


Fig. 36—Curved beam under uniform bending. Positive direction of moment is shown

boundary conditions to be met by the stress system on the curved edges are

A) When $\rho = r_1$ or r_2, $S_\rho = 0$

The stress being independent of θ, the end conditions must be met by every section. Therefore

B) $v_{\rho\theta} = 0$

C) $\int_{r_1}^{r_2} S_\theta (b\, d\rho) = 0$

D) $\int_{r_1}^{r_2} \rho S_\theta (b\, d\rho) = -M$

Equation 68, 69, and 70 (from the preceding section) are the general equations for ring distribution and are

$$S_\rho = \frac{B}{\rho^2} + 2C + D(1 + \log_e \rho^2) \quad \text{.......... (68)}$$

$$S_\theta = -\frac{B}{\rho^2} + 2C + D(3 + \log_e \rho^2) \quad \text{.......... (69)}$$

$$v_{\rho\theta} = 0 \quad \text{.......... (70)}$$

Condition (B) is satisfied by Equation 70. Substitution and integration shows that (A) includes condition (C). The constants in the stress equations may then be determined from (A) and (D). Solving and substituting gives[1]

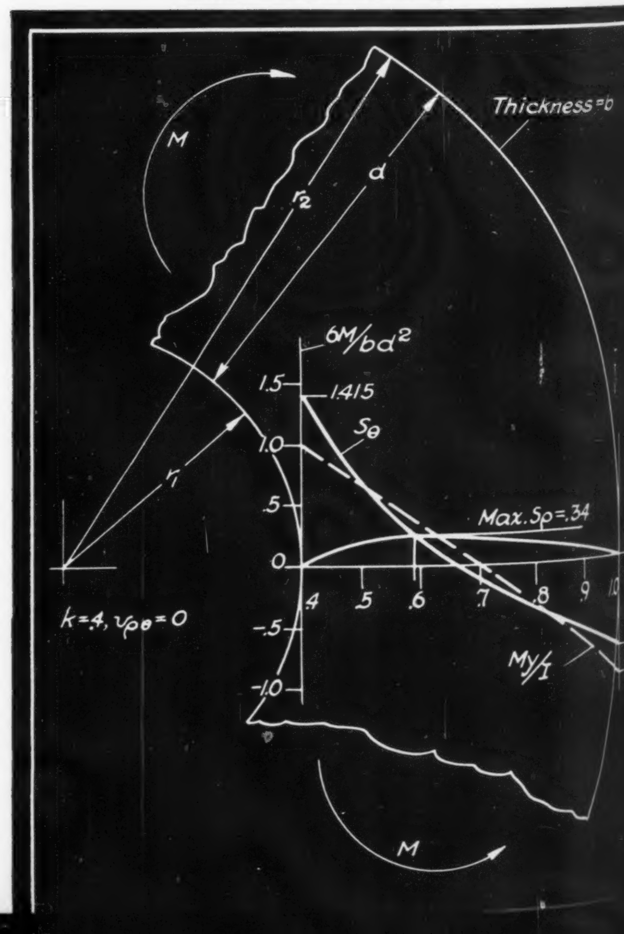
$$S_\rho = \frac{6M}{bd^2} K_m \left[\left(1 - \frac{r_1^2}{\rho^2}\right) m - (1-k^2) \log_e \frac{\rho}{r_1} \right] \quad (92)$$

$$S_\theta = \frac{6M}{bd^2} K_m \left[\left(1 + \frac{r_1^2}{\rho^2}\right) m - (1-k^2) \left(1 + \log_e \frac{\rho}{r_1}\right) \right] \quad \text{.......... (93)}$$

where k is the radius ratio r_1/r_2, $m = -\log_e k$, and the factor K_m is given by

$$K_m = \frac{2(1-k)^2}{3[(1-k^2)^2 - (2mk)^2]} \quad \text{.......... (94)}$$

Fig. 37—Stress over cross section of a curved beam under uniform bending



[1] Original solution of this problem and of the one that follows, on shear loading, is credited to H. Golovin, St. Petersburg, 1881.

ractical Design

This factor, K_m, will always be positive as k varies through its range from 0 to 1. For positive moment, S_ρ is always tension. S_θ is tension and a maximum at r_1, compression at r_2. The stress variation over the cross section of a typical beam is shown in *Fig.* 37. The straight beam formula, My/I, is plotted for comparison.

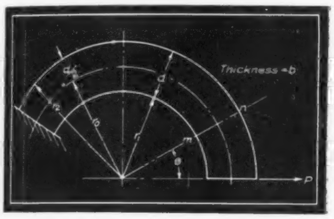


Fig. 38 — Curved beam under a radial shear load. Positive direction of P and Θ shown

The stress on the curved boundaries is obtained by substituting ρ equals r_1 or r_2 in Equation 93

$$S_\theta = \frac{6M}{bd^2} K_1, \qquad \rho = r_1 \dots\dots\dots\dots (95)$$

$$S_\theta = -\frac{6M}{bd^2} K_2, \qquad \rho = r_2 \dots\dots\dots\dots (96)$$

where

$$K_1 = [2m - (1-k^2)]K_m$$

and

$$K_2 = [(1-k^2) - 2mk^2]K_m$$

Since the stress at the boundaries of a straight beam are given by

$$S = \pm \frac{Mc}{I} = \pm \frac{6M}{bd^2}$$

K_1 and K_2 may be considered correction factors to be applied to the straight beam for obtaining the curved beam stress.

Correction factors K_1 and K_2 are plotted against k in *Fig.* 39. The other data on this chart will be developed later. The analysis of curved beams under uniform bending may be made easily with the aid of these curves. For values of k greater than .85 (r_1 approximately 6 times d) the correction to the straight beam is less than 5 per cent. As k approaches 1, with d fixed, the curved beam becomes straight and the stress Equations 95 and 96 reduce to the straight beam equation.

As k approaches zero the stress at r_1 increases without limit. For k less than .05, K_1 is closely approximated by

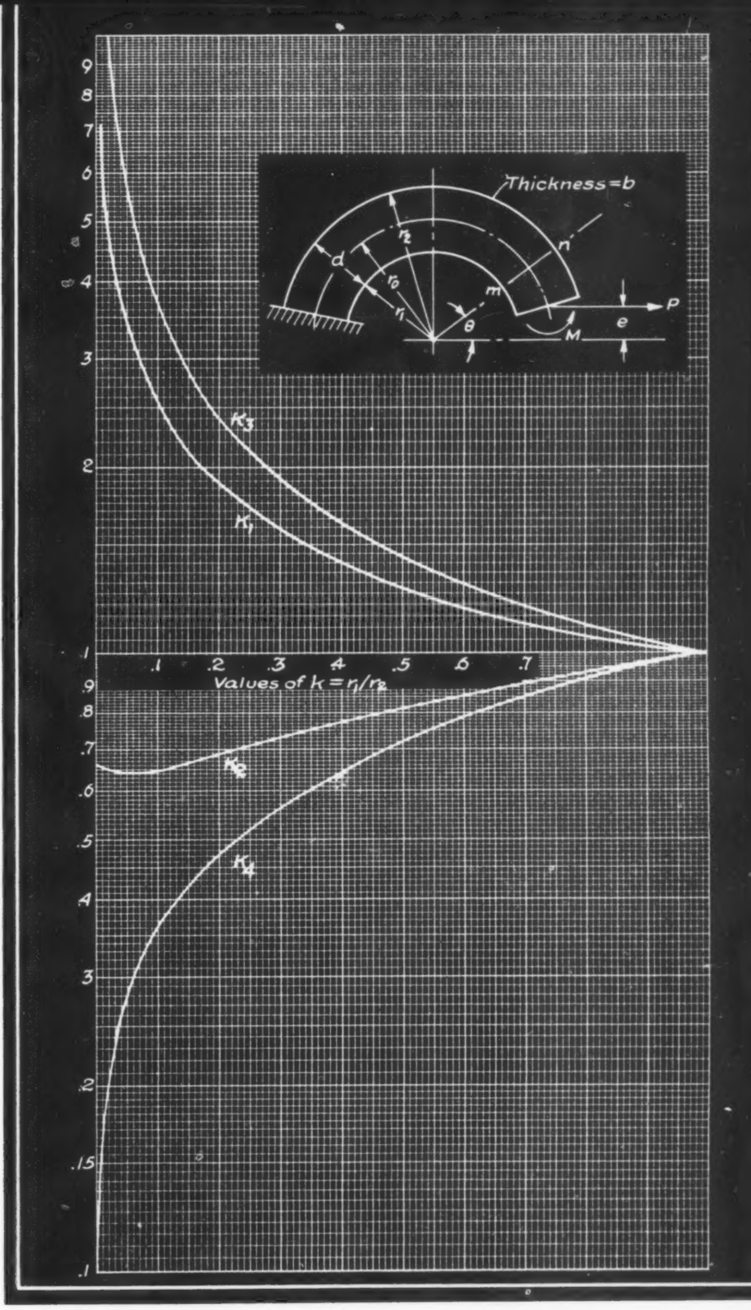


Fig. 39—Boundary stress in curved beams under moments and concentrated loads. Positive directions of forces and distances are shown

$$K_1 = \frac{2}{3}(2m-1), \quad k < .05 \dots\dots\dots\dots (97)$$

K_2 approaches the limit of 2/3.

Illustrated in *Fig.* 38 is a curved beam under a radially directed concentrated load. The combination of the stress system from this and the uniform bending load of *Fig.* 36 will permit evaluating the stress for any loading. The stress system from this loading is developed from the stress function

$$\phi(\rho, \theta) = \left(\frac{A_1}{\rho} + B_1\rho^3 + C_1\rho + D_1\rho\log_e\rho \right) \sin\theta$$

Substitution in Equation 66 (Part IV), the compatibility equation for polar co-ordinates, shows that the stress system obtained from this function is permissible. Substitution in Equations 63, 64,

and 65 gives for this stress system

$$S_\rho = \left(-\frac{2A_1}{\rho^3} + 2B_1\rho + \frac{D_1}{\rho} \right) \sin\theta$$

$$S_\theta = \left(\frac{2A_1}{\rho^3} + 6B_1\rho + \frac{D_1}{\rho} \right) \sin\theta$$

$$v_{\rho\theta} = -\left(-\frac{2A_1}{\rho^3} + 2B_1\rho + \frac{D_1}{\rho} \right) \cos\theta$$

From *Fig.* 38 the boundary conditions to be met are

A) When $\rho = r_1$ or r_2, $S_\rho = v_{\rho\theta} = 0$

B) When $\theta = 0$, $S_\theta = 0$

C) When $\theta = 0$, $\int_{r_1}^{r^2} v_{\rho\theta}(b\,d\rho) = -P$

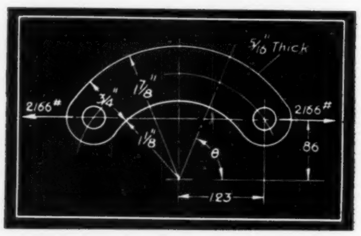


Fig. 40 — Curved link under tension

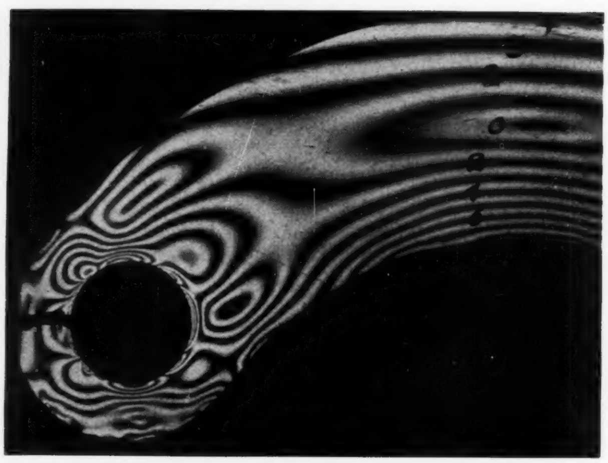


Fig. 41—Photoelastic picture of link in Fig. 40. Enlargement is 2.2 diameters

From these conditions the constants A_1, B_1, and D_1 are found, giving for the stresses

$$S_\rho = \frac{P\sin\theta}{bd} K_p \left[-\left(\frac{r_1}{\rho}\right)^3 - k^2\left(\frac{\rho}{r_1}\right) + (1+k^2)\left(\frac{r_1}{\rho}\right) \right] \ldots\ldots (98)$$

$$S_\theta = \frac{P\sin\theta}{bd} K_p \left[\left(\frac{r_1}{\rho}\right)^3 - 3k^2\left(\frac{\rho}{r_1}\right) + (1+k^2)\left(\frac{r_1}{\rho}\right) \right] \ldots\ldots (99)$$

$$v_{\rho\theta} = -\frac{P\cos\theta}{bd} K_p \left[-\left(\frac{r_1}{\rho}\right)^3 - k^2\left(\frac{\rho}{r_1}\right) + (1+k^2)\left(\frac{r_1}{\rho}\right) \right. \ldots\ldots (100)$$

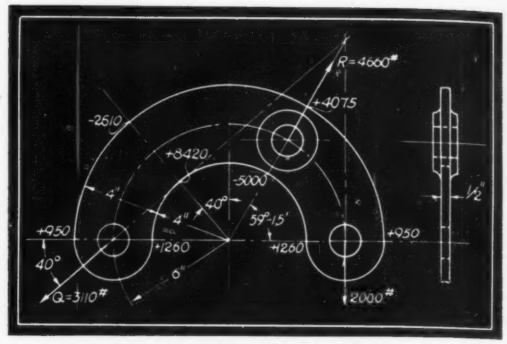


Fig. 42—Curved rocker link. Stress at maximum and minimum points are shown

where the stress constant K_p is given by

$$K_p = \frac{1-k}{k\,[(1+k^2)m - (1-k^2)]}$$

and, as before, $k = r_1/r_2$ and $m = -\log_e k$.

Stress constant K_p will always be positive. S_ρ is always tension (for positive direction of P) and a maximum near the middle, as is also $v_{\rho\theta}$. S_θ is compression at r_2, tension and a numerical maximum at r_1. The principal directions are radial and tangential only at $\theta = \pi/2$ and at $\rho = r_1$ or r_2. At $\theta = 0$, $S_\rho = S_\theta = 0$ and the stress is pure shear.

Considering Beam a Section of Ring

By Saint Venant's principle the beam need not start at $\theta = 0$, but may start at some other section, say *m-n*, provided the loading on *m-n* is statically equivalent to *P*, (in both amount and location) and the stresses close to *m-n* are not required. The requirement then is that the resultant of the loading be applied through the center of curvature. θ is measured from the direction of *P*.

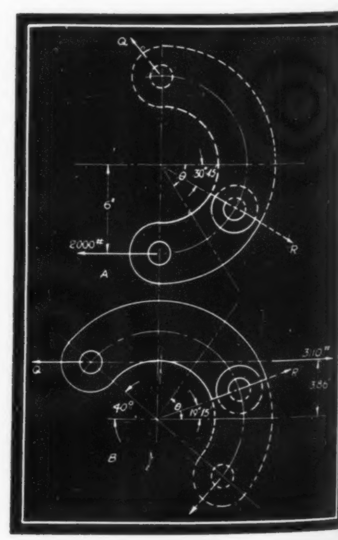


Fig. 43 — Rocker link shown in Fig. 42. Analysis of portions are shown in solid lines

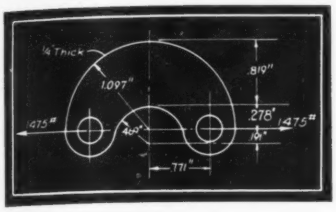


Fig. 44 — Curved link with eccentric radii

The surface stress S_θ is obtained by substituting $\rho = r_1$ or r_2 in Equation 99

$$S_\theta = \left(\frac{6\, Pr_0 \sin\theta}{bd^2}\right) K_3, \qquad \rho = r_1 \ldots\ldots\ldots\ldots (101)$$

$$S_\theta = -\left(\frac{6\, Pr_0 \sin\theta}{bd^2}\right) K_4, \qquad \rho = r_2 \ldots\ldots\ldots\ldots (102)$$

where

$$K_3 = \frac{2}{3}(1-k)^3 K_p$$

and

$$K_4 = \frac{2}{3} k\, (1-k)^2 K_p$$

and r_0 is the radius of the center line of the beam $= (r_1 + r_2)/2$. In *Fig.* 39 K_3 and K_4 are plotted against k.

Chart Simplifies Calculations

The stress from any loading may now be found by combining the stress systems, developed for the loadings of *Figs.* 36 and 38. If the load is applied entirely along one radial section it may be reduced to a single load P acting through the center of curvature, and a moment M. The stresses at any point from P and M acting separately may be found from Equations 92, 93, 98, 99, and 100, and then combined algebraically to give the resultant stress. This will, of course, apply only to stresses beyond the loaded section.

If there is a moment on the section in addition to the other loads it is to be added to the moment obtained by transferring the resultant to the center of curvature.

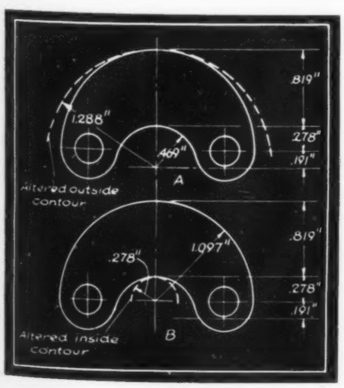


Fig. 45—Analysis of link in Fig. 44 by modification of contour at A and B to obtain stress on inside and outside radius respectively

The above process is illustrated by the diagram at the top of *Fig.* 39. P is the resultant load on the end section and e the "eccentricity" of the load as regards the center of curvature. The moment, after transferring P to the center, is then M-Pe. The above process may be combined into a single expression for the stress at r_1 and r_2, thus

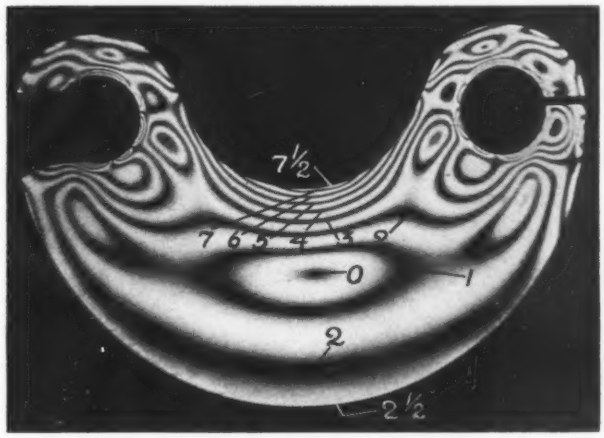
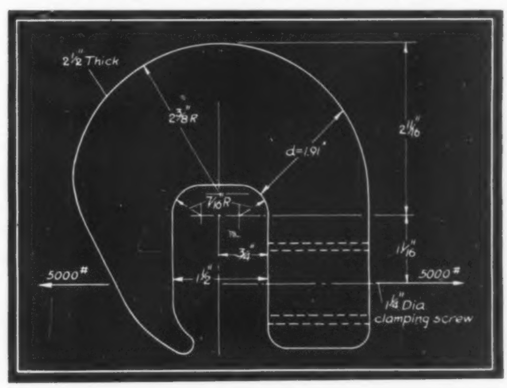


Fig. 46—Photoelastic picture of link of Figs. 44 and 45. Enlargement is 1.4 diameters

Fig. 47—Stress in throat fillets of C-clamp is calculated by theory of curved beams



$$S_\theta = \frac{6}{bd^2}\left[K_1 (M - Pe) + K_3\, Pr_0 \sin\theta\right], \qquad \rho = r_1 \,(103)$$

$$S_\theta = -\frac{6}{bd^2}\left[K_2 (M - Pe) + K_4\, Pr_0 \sin\theta\right], \qquad \rho = r_2 \,(104)$$

as noted on *Fig.* 39. The stress constants K_1, etc., are taken from the diagram, making the solution of curved beams no more involved than that of a straight beam.

Where loads are applied to several sections the analysis is begun with the first loaded section, the

(*Continued on Page* 134)

Strength and resistance to abuse in Welch air circulator (Right) is achieved by the leg design and supporting frame without detracting in any way from appearance. Styled by Alfred F. Fukal, protective grill results in a definite increase in circulated air volume. Unit is finished in two-tone opalescent gray or two-tone statuary bronze

Claimed to be more efficient than gun type burners, a swirling, diffused oil flame is used in the Conco air conditioner (Left). Single heavy-duty motor drives both burner and blower, the whole assembly being mounted on rubber to isolate vibration and reduce noise. Trouble-free life is assured by use of single stage, positive displacement, rotary fuel pump

Desig

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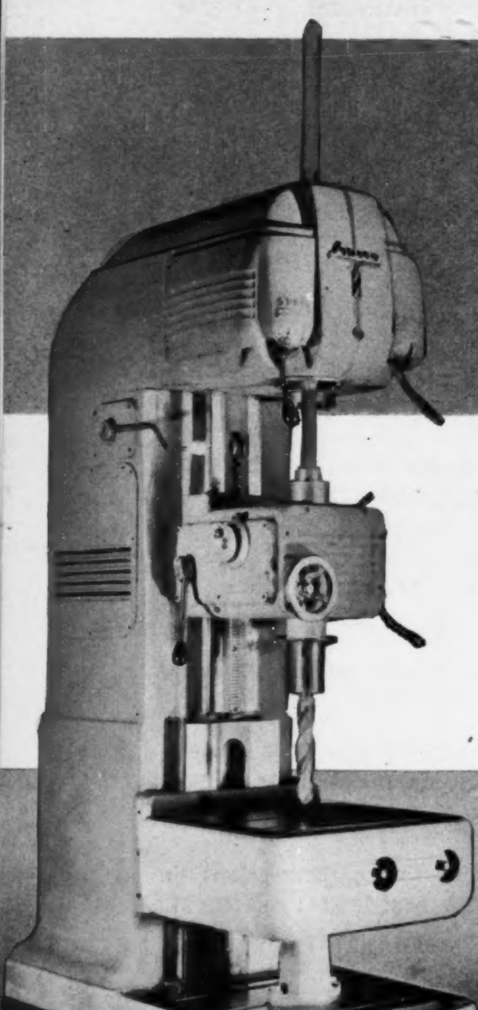
(For new mach... page 15

THIS MONTH'S COV... provides refrigeration... For maximum efficiency ... and condenser. The latter ... construction type. Conden... revaporized by the hot cond... as a vapor. Circulating fan... are operated by a separate ... refrigerating compressor. ... ings, this Philco unit insu...

uperheterodyne radio (Above), incorporating four low-drain tubes, has an attractively styled case molded of tan onyx tenite. Trim is of matching leatherette. Careful attention to styling is evidenced by rounded corners and blended contours of the case. Freedom from damage to the instrument panel and fluted speaker grill of this Sentinel radio, as a result of inadvertent handling, is assured by the provision of a door. Control switch recessed in housing prevents unintentional operation



Precision as well as sturdy construction has been built into the 25-inch swing drill (Left). Drive of this Sibley press is by means of heat treated, alloy steel gears mounted on horizontal multisplined shafts. Transmission is of the removable unit-type with all shafts carried in antifriction bearings for permanent accuracy

Incorporating three advanced design features, the American laundry tumbler (Right) uses a single electric motor with an extended shaft to operate the blower and to revolve the cylinder. Paddle-wheel type of blower draws air over steam coils, directing it diagonally through the work zone. Equivalent of one-piece drum construction is attained by use of spot welded galvannealed steel sheet

Containing over fifty individual parts molded of bakelite, transmission of vibration in the Prince Castle counter mixer (Below) is prevented by mounting the base on rubber. Maintenance of attractive appearance is assured by making all exposed metal parts of stainless steel. Each mixer unit is driven from the single motor by means of an individual clutch. Shielded bakelite spindle bearings eliminate necessity for petroleum lubricants

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S COV l airconditioning unit
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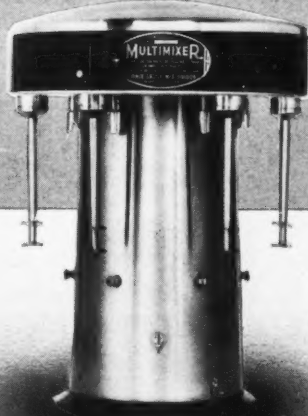
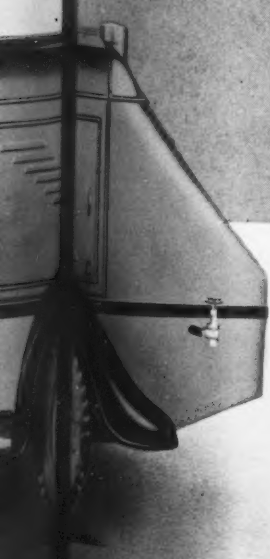


Spindle drive is accomplished through belts from a two-horsepower, fan-cooled, ball bearing, fully enclosed motor on Carboloy tool grinder (Right). To facilitate adjustment when handling large tools, the work rest tables are individually manipulated by means of screws operated by crank handles



Capable of producing 200 pounds of steam in two minutes, the machine illustrated (Left) is designed for the purpose of heating asphalt, tar, cutback, road oils, etc., for unloading these materials from tank cars. Developed by Littleford Bros., staggered coil assembly is utilized in such a manner that combustion gases completely envelop each coil. Boiler is oil fired. Circulation of boiler water is achieved by use of a 1½-horsepower gasoline engine

MACHINE *Editorial* DESIGN

New Designs Should Be Restricted to Most Pressing Needs

GRAVELY impending events may well make desirable a much stricter limitation in development of machines for purely domestic purposes. Such a situation has of course already arisen insofar as the automobile industry is concerned, in which a twenty per cent curtailment in production of cars has gone into effect. That this figure is likely to be increased and that the same trend will reach into other industries is becoming clearly evident.

Apparent also is the dire need for reduction in radical changes and new models except in those classes of machines and equipment directly or indirectly allied with the defense program. Again citing the automobile industry, it is believed that refinements only will be incorporated in 1942 models which will outdate this year's cars almost as effectively as complete changeovers in design. These refinements may be necessitated by shortage of materials such as nickel and zinc, and thus serve not only to give the effect of new models but also to conserve strategic materials.

Assuming that many other mass production industries—taking as possible examples the refrigerator, radio and domestic cleaner—are affected by curtailment in development and production, the immediate effect of the release of designers and shop men from such work can readily be seen. The biggest job we have ever done is ahead of us, and every ounce of manpower and energy should be sought that can be converted to its effective accomplishment. The effort that under ordinary circumstances would be put into design and manufacture of consumer goods over the next few years obviously should be utilized in assuring that markets will remain to us for those classes of goods when the "shooting" is over!

Of untold advantage to the country when that time eventually comes will be the huge backlog of orders or potential orders in all the industries that voluntarily or by request cut down their production during the period of strife. Many economists and business leaders already are visualizing the probable letdown when the need for an enforced defense program ceases. What better assurance could there be of immediate demand, employment and regenerative activity at that time than an absolute maximum limitation on nonessential production during the war?

Professional Viewpoints

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

" . . . can enter both contests"

To the Editor:

I have noted with interest the announcement of the Summerill Tubing Co., Bridgeport, Pa., discussing the establishment of a series of prizes for papers to advance the art of welding of aircraft steels including tubing and other parts for tubular assemblies.

My purpose in writing you is to point out that papers submitted in the Summerill contest will also be eligible in the $200,000 Industrial Progress Award Program which is being sponsored by the James F. Lincoln Arc Welding Foundation, provided of course, that such papers comply with the Foundation's Rules and Conditions as regards subject matter and form.

It is conceivable that a paper entered in the Summerill contest would be written in such a way as to be acceptable in the Lincoln program. If it is not acceptable as written, the author, by a few changes or additions, could make it conform to the Foundation requirements.

I felt this would be of interest since eligibility in the two projects should provide added stimulus to the study of welding in the aircraft field.

The Summerill Tubing Co. is in full accord with this plan.

—ED C. POWERS
The James F. Lincoln Arc Welding Foundation

"...high frequency power for aircraft?"

To the Editor:

Referring to the much discussed subject of the use of high frequency alternating current on aircraft, many interesting possibilities and advantages over direct-current equipment are introduced. At the present time, one difficulty prevents the general adoption of alternating current to aircraft with large electric power requirements—that of providing a lightweight constant-speed alternator drive powered by the main engines.

Experience has shown that the use of a constant speed auxiliary power plant to drive an alternator is inefficient from a weight standpoint because of the reduction in horsepower from the alternator prime mover with increase in altitude and the complication of supercharging to reduce the effect of this horsepower decrease.

Present design trends indicate that more efficient supply of alternating current on an airplane would be obtained by utilizing a lightweight constant-speed driving device to couple the alternator to the main engine. This device would maintain constant r.p.m. on the alternator, despite changes in main engine speed during operation of the airplane. Development is now in progress to produce a device of this type. Such a constant-speed device is necessary to maintain a constant-frequency output from the alternators.

In event a suitable, constant-speed source for driving alternators from the main engine of an airplane is developed, numerous advantages would occur through the use of an alternating current system:

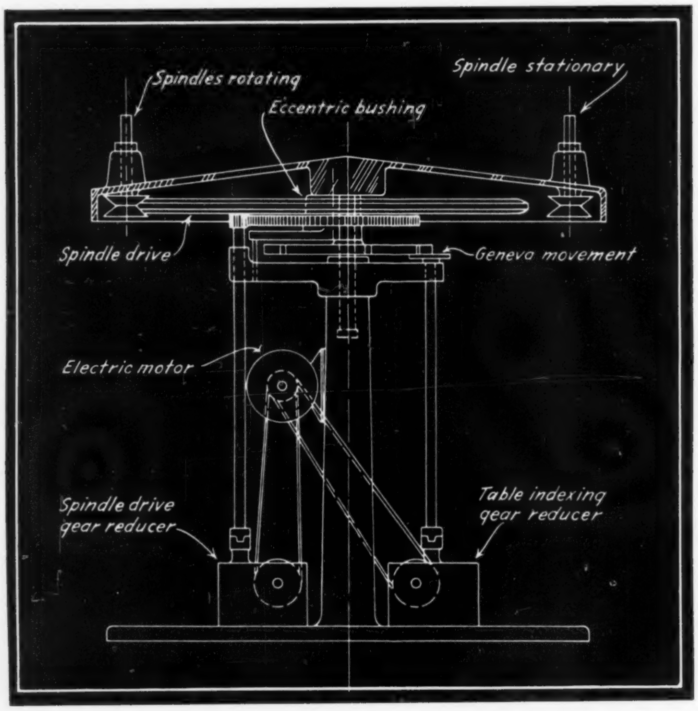
1. Ability to transform voltages with transformers
2. Use of squirrel-cage induction motors for auxiliary equipment, eliminating commutators and slip rings
3. High operating speeds allow low weight
4. Possible fire hazards would be greatly reduced because no arcing can occur in motors as at the brushes of direct-current equipment
5. High voltages allow the use of smaller wires, reducing the total weight of the installed electrical system
6. Alternating current equipment, by obviating commutators, will allow motors and alternator to operate at high altitudes and avoid difficulties which would occur on high voltage direct-current equipment due to lower ionization voltages which occur at low air pressures of high altitudes
7. At the present time, there is considerable controversy as to whether single-phase, 800-cycle,

(*Concluded on Page* 122)

Applications
of Engineering Parts and Materials

Dual Drive Feeds Sandblaster

NOTEWORTHY by reason of the ingenious but simple solution of its feed table drive problem is an automatic sandblast machine made by Leiman Bros. Inc. Machine is equipped with four complete sandblast units, work being

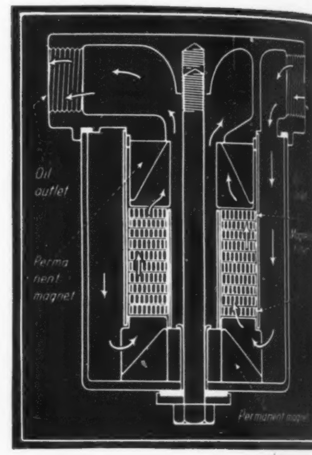


fed to each by means of the table mechanism shown in the sketch.

The mechanism illustrates a somewhat unusual application of a single electric motor driving—via V-belts—two gear reducers, one of which rotates the spindles while the other drives the table through a geneva mechanism. Spindle-rotating drive consists of a friction wheel mounted eccentric to the table in such a manner that rotation occurs only while the piece is being processed. Geneva stop mechanism provides an intermittent table rotation for processing the work and loading and unloading the spindles.

Magnets Filter Metal Particles

POWERFUL permanent magnets inducing magnetism in a stack of screens are used to remove particles of iron or steel from lubricating oil circuits. Developed by S. G. Frantz Co. Inc., the filter effectively extracts particles as fine as one micron, thereby providing adequate protection for machine bearing parts.

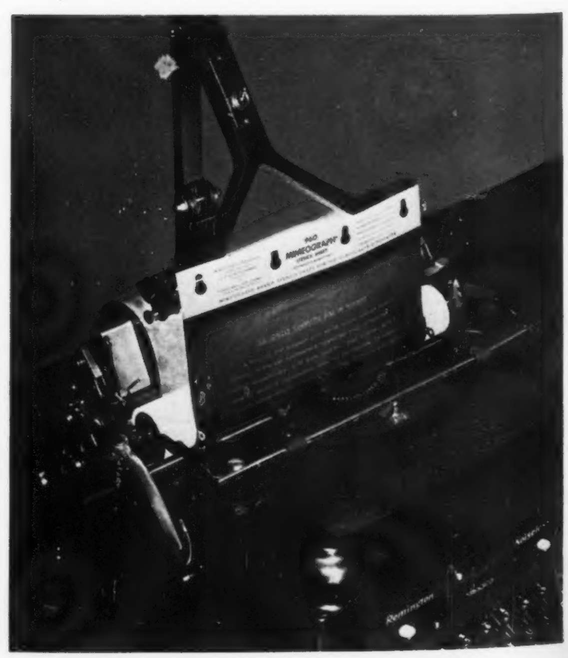


In operation it has been found that many substances including lead and other nonmagnetic materials have been collected, probably because such microscopic particles have some affinity for iron and are removed with it. Similar filters are available powered by electromagnets.

Plastic Improves Typewriter Platen

TRANSPARENT typewriter platen molded of DuPont Lucite is instrumental in eliminating errors in the cutting of stencils. Platen contains a built-in, six-watt fluorescent light which makes the stencilled letters clearly visible as they are cut.

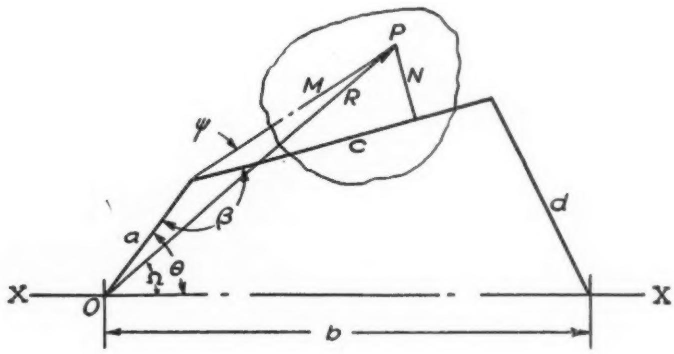
Introduced by the Lumirol Co., the platen, because of the combination of hardness and resilience of the resin, is claimed to have a practically unlimited life and to be free from the possibility of marring by the typewriter keys.



Part III—Characteristics of a Point on Connecting Link

By Guy J. Talbourdet

Research Division
United Shoe Machinery Corporation



Parts I and II of these Engineering Data Sheets which appeared in the May issue concerned the mathematical determination of the motion characteristics of the cranks of 4-bar linkages. Knowing the instantaneous displacement and angular velocity of one crank in such a linkage, this extension enables the engineer to determine complete velocity and acceleration data for any point on the connecting link. With these three parts as a basis, Part IV, which will appear in the July issue, deals with the balancing of the 4-bar linkage.

Calculate the values of the following factors from the geometry of the linkage:

$$A = a \sin \theta$$

$$B = a^2 + b^2 - 2\,ab \cos \theta$$

$$E = a - b \cos \theta$$

$$H = T - 2\,ab \cos\theta$$

$$T = a^2 + b^2 + c^2 - d^2$$

$$\beta = \pm \psi + \cos^{-1}\frac{E}{\sqrt{B}} + \cos^{-1}\frac{H}{2c\sqrt{B}}$$

$$\text{where } \psi = \sin^{-1}\frac{N}{M}$$

ψ = angle between link c and line M; positive when P is above link c and negative when P is below link c

M = length of line drawn from point P to intersection of crank a and link c

N = perpendicular distance from P to link c

V_R and a_R = linear velocity and acceleration respectively of point P along radius vector R

ω_P and α_P = angular velocity and acceleration respectively of point P about center O

ω = angular velocity of crank a

Then

Length of radius vector $OP = R = \sqrt{a^2 + M^2 - 2a\,M\cos\beta}$

Angle between radius vector OP and $X-X$ axis $\Omega = \theta - \cos^{-1}\dfrac{a - M\cos\beta}{R}$

Knowing R and Ω, the position of point P with respect to center O and $X-X$ axis is determined.

Therefore

$$V_R = \frac{dR}{dt} = \frac{aM\omega\sin\beta}{R}\left[\frac{a\,E}{B} - \frac{Ab\left(2-\dfrac{H}{B}\right)}{\sqrt{4\,c^2\,B - H^2}} - 1\right]$$

$$\omega_P = \frac{d\Omega}{dt} = \omega + \frac{\omega(M^2 - aM\cos\beta)}{R^2}\left[\frac{aE}{B} - \frac{bA\left(2-\dfrac{H}{B}\right)}{\sqrt{4\,c^2\,B - H^2}} - 1\right]$$

and

$$a_R = \frac{d^2R}{dt^2} = \frac{abM\omega^2\sin\beta}{R}\left[\frac{A}{B} - \frac{2aEA}{B^2} - \frac{2bA^2\,(c^2 - d^2)}{B^2\sqrt{4\,c^2\,B - H^2}}\right.$$

$$\left. - \left(2 - \frac{H}{B}\right)\left(\frac{a\cos\theta}{\sqrt{4c^2B - H^2}} - \frac{2\,bA^2\,(2\,c^2 - H)}{(\sqrt{4c^2\,B - H^2})^3}\right)\right]$$

$$+\left[\frac{a\,E}{B} - \frac{b\,A\left(2-\dfrac{H}{B}\right)}{\sqrt{4\,c^2\,B - H^2}} - 1\right]^2\left[\frac{a\,M\,\omega^2\cos\beta}{R} - \frac{a^2\,M^2\,\omega^2\sin^2\beta}{R^3}\right]$$

$$a_P = \frac{d^2\Omega}{dt^2} = \frac{b\omega^2\,(M^2 - aM\cos\beta)}{R^2}\left[\frac{A}{B} - \frac{2\,a\,EA}{B^2} - \frac{2bA^2\,(c^2 - d^2)}{B^2\sqrt{4\,c^2\,B - H^2}}\right.$$

$$\left. - \left(2 - \frac{H}{B}\right)\left(\frac{a\cos\theta}{\sqrt{4c^2\,B - H^2}} - \frac{2b\,A^2\,(2c^2 - H)}{(\sqrt{4c^2\,B - H^2})^3}\right)\right]$$

$$+\frac{a\,M\omega^2\sin\beta\,(a^2 - M^2)}{R^4}\left[\frac{aE}{B} - \frac{bA\left(2-\dfrac{H}{B}\right)}{\sqrt{4c^2\,B - H^2}} - 1\right]^2$$

It is therefore apparent, knowing: (1) The linear velocity of point P in the direction of radius vector R, V_R; (2) the angular velocity, ω_P, of radius vector R, about its center O; and (3) the length of radius vector R, that the resultant instantaneous velocity may be determined.

The product of ω_P, and R yields the velocity of point P in a direction normal to R. Resolution of this quantity with V_R will give the absolute velocity of the point P with respect to the system.

Similar treatment applied to the linear acceleration a_R and the angular acceleration a_P will give the resultant acceleration of point P with respect to the system.

ASSETS *to a* BOOKCASE

Mechanisms

By Stanton E. Winston, Associate Professor of Mechanical Engineering, Illinois Institute of Technology; published by American Technical Society, Chicago; 372 *pages,* 5½ *by* 8¼ *inches, clothbound; available through* MACHINE DESIGN *for* $3.50 *postpaid.*

A fundamental discussion of the theory of transmission of motion and mechanical modifications therefore, its treatment is based on practical explanations for the understanding of engineering factors involved. Each mathematical principle is developed by actually showing the complete calculations. Likewise formulas are developed by substituting for known conditions and solving.

Illustrative examples are presented for each principle to show practical applications. In addition, over 250 practice problems are included to aid a student in grasping the theory involved in practical design work. Constrained motion, links, driver and follower, higher and lower pairing, turning and sliding pairs, incomplete pairing, graphic solutions, instantaneous motion, velocity diagrams, trains of mechanisms, and cams are some of the subjects covered.

□ □ □

Engineers' Manual of Statistical Methods

By Leslie E. Simon; published by John Wiley & Sons, Inc., New York; 231 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN *for* $2.75 *postpaid.*

This useful book discusses the important subject of statistics as applied to practical problems using logical, commonsense methods. It treats those phases of the science of statistical method which are needed by an engineer in making analyses of data involving sampling, probability, etc.

Presentation is concise and controversial matters are neglected except insofar as their discussion is necessary to the development of the important phases of the subject. Extended explanations and methods too important to be omitted altogether are treated briefly or in appendices.

Covered are relations between sample and lot, with methods for checking their relationship; ranges of accuracy for size of samples; methods of inspection; measurement of quality by failures; probability with respect to significant differences; statistical techniques. Particularly useful are charts in a cover pocket for determining ranges of error and for reading the inverse solution of the incomplete beta-function ratios, saving considerable time over the use of unwieldy tables.

□ □ □

Trade-Names Index

Published by the Special Libraries Association, New York; 178 *pages,* 7 *by* 10 *inches, clothbound; available through* MACHINE DESIGN *for* $4.00 *postpaid.*

Index contains definitions, sources and a bibliography of tradenames and trademarks. The list is carefully compiled from the files of the technology department of the Carnegie Library of Pittsburgh. A supplementary classified bibliography of sources for tradenames, trademarks, and brandnames is included to aid the user who does not find a tradename in the main text.

The volume contains an alphabetical list of 3496 tradenames. For each a definition is given and usually a reference is cited for further information. In general, the list is concerned with materials, processes, and equipment having technical significance.

□ □ □

Writing the Technical Report

By J. R. Nelson, Professor of English, College of Engineering, University of Michigan; 373 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN *for* $2.50 *postpaid.*

How to solve problems of technical report writing and present a complete, understandable treatise is the purpose of this authoritative book. It tells how to analyze the type of report, how to choose the best form and style, how to organize the material, how to use figures, tables and annotations.

It is especially useful to young engineers who have been limited in experience with report writing. The author stresses how improved reports can eliminate misunderstanding, save time in explanations, demonstrate the writer's knowledge and ability, and create favorable impressions on employers.

Also treated as indispensable are that the writer take the reader's point of view and that he keep in mind, at all times while writing, the one objective of the report and the definite group of readers he is addressing.

New PARTS AND MATERIALS

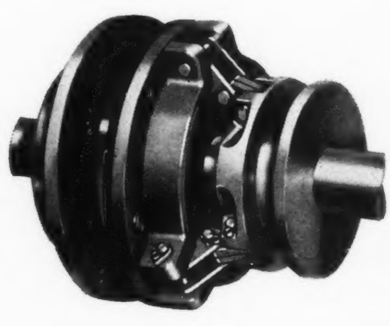
Coupling Dampens Peak Loads

ESPECIALLY designed for connecting diesel, oil, gasoline or gas engines to any type of driven machinery as well as for use with motor-driven compressors, single-cylinder pumps, etc., a new type of flexible coupling known as "Airflex" has been announced by The Falk Corp., Milwaukee. The coupling consists of a special resilient rubber gland, lined with a premoulded, prevulcanized cylindrical section treated to prevent air loss. This cylindrical lining is covered with multiple alternating layers of durable fabric and live rubber, the entire gland being permanently bonded to rugged steel inner and outer

rims. Through a standard thread valve which will take any hand pump fitting or air nozzle, the gland is inflated to proper pressure. By changing air pressure, the degree of elasticity in coupling can be varied. The coupling is available in six standard types: Type AR, both hubs external, for new installations where gap between shaft ends equivalent to width of coupling gland can be provided; Type AR, one hub inverted, for use where gap between shaft ends is small or where space is restricted; Type FM, designed for equipment having flywheel on driving or driven shaft; Type AR, with thrust unit, for marine propulsion applications and to transmit thrust through coupling to an adjacent thrust bearing; Type XR, high speed type, for applications where speed is beyond the allowable for the basic AR type; and the disconnecting coupling for dual drives and other requirements where quick disconnecting is required.

Toggle-Type Clutch Easily Installed

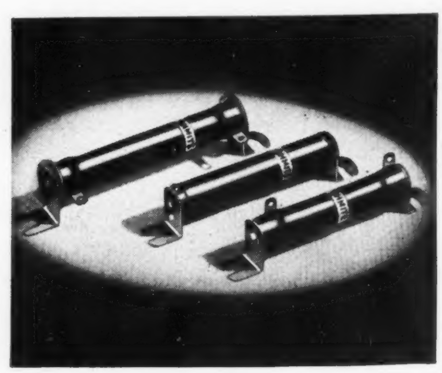
TOGGLE-TYPE, over-center clutch has been developed by Rockford Drilling Machine Division, Borg-Warner Corp., 1315 Eighteenth avenue, Rockford, Ill., for use with gasoline or other motors and in applications up to six horsepower. Known as Type LMC the clutch has operating links, pins, rollers and clutch plates of hardened steel. Pressure for driving is uniformly distributed and applied directly opposite the facing, where it is most effec-



tive. No oiling is necessary for the metal brushing which carries driven sprocket, pulley, sheave, flange or other part of machine in which clutch is installed. The clutch transmits 1 horsepower at 100 revolutions per minute, and proportionately smaller or larger horsepower as speed decreases or increases, up to a maximum of 6 horsepower.

Offer "Live" and "Dead" Resistors

WIRE-WOUND, sealtight, vitreous-enameled resistors, manufactured by Ohmite Mfg. Co., Dept. 10, 4835 Flournoy street, Chicago, are now available in "live" and "dead" bracket types for special applications. The "live" type resistor has flexible leads connected to tin-plated brass brackets,

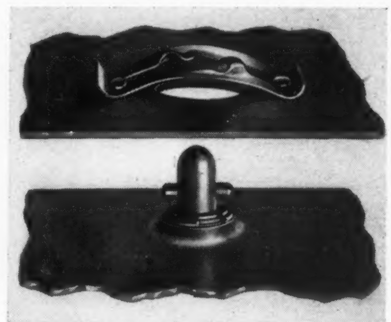


and is designed for mounting and making electrical connection by bolting slotted brackets to panel terminals. The "dead" type is mounted by bolting to brackets. Electrical connections are made separately to lugs. Brackets for one, two or three re-

sistors are mounted to resistors by through-bolts. Leakage distance from lug to bracket can be regulated by use of mica washers, or by having lugs located as far in as required. Available in a wide range of core sizes with diameter from 9/16 to 2½ inches, both can be used for signal circuits, electrical refrigeration controls, switchboard and other applications.

Cowl Fastener for Sheet Metal

FOR fastening engine cowlings, handhole covers, access plates and similar parts, United-Carr Fastener Corp., Cambridge, Mass., has developed the Dot Cowl fastener which consists of a stud, ring, spring and rivet. The self-aligning fastener may be engaged or disengaged readily on flat or curved sur-

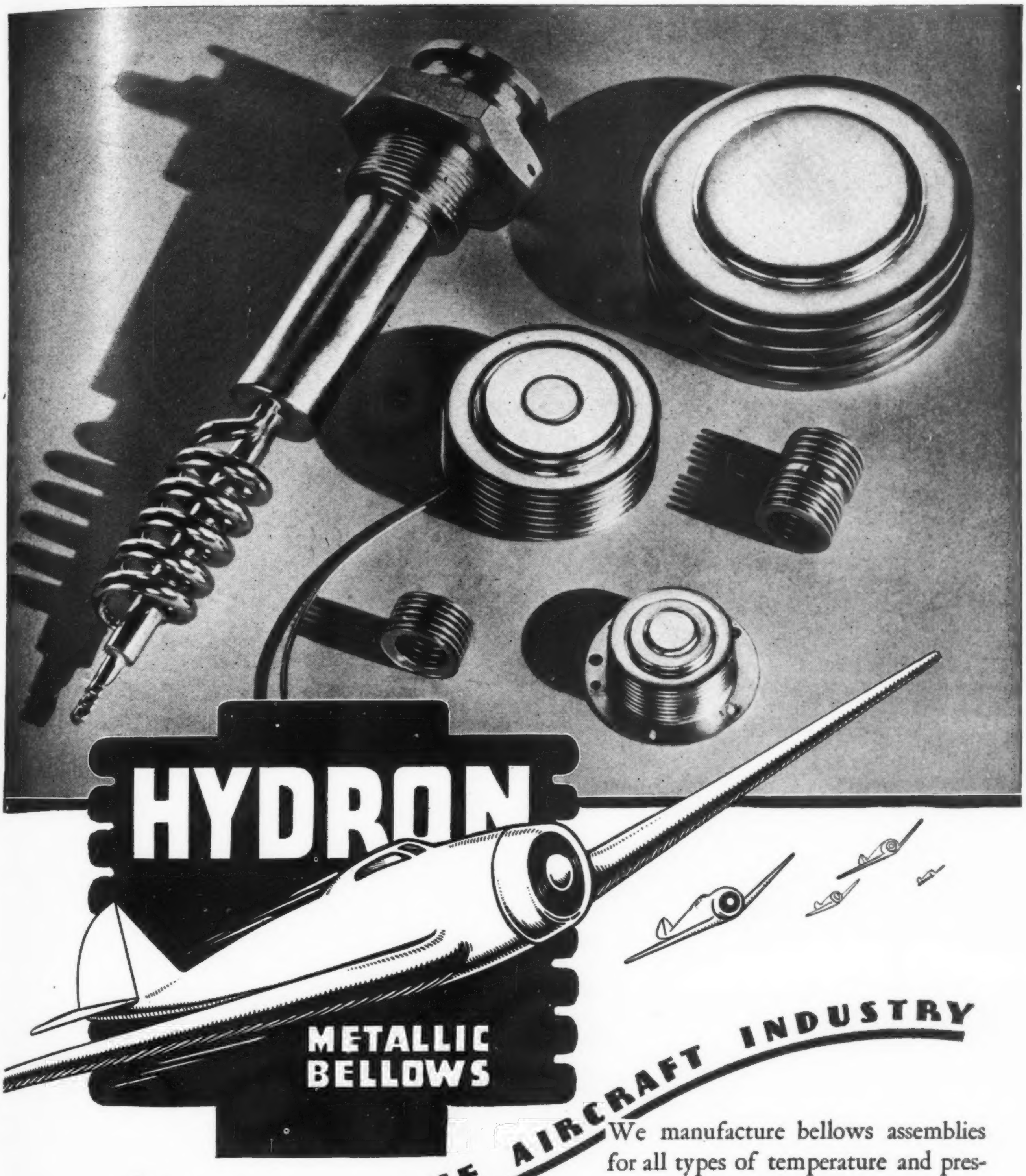


faces. When the stud is turned counterclockwise, the pin working on the under surface side of the cam acts as an ejector. The stud pin, in other words, may pass under the spring during detachment to force plates apart quickly. The fastener has approximately 1/16-inch pick-up or lift, and is adapted to variations of at least 1/32-inch in thickness without change of location of pin. Spring-hardened and cadmium-plated to resist rust, corrosion and wear, it may easily be installed or removed without risk of losing any part of the new self-aligning fastening assembly.

Aircraft Motor Is Lightweight

EXPRESSLY designed for applications in the aircraft industry, the new fractional horsepower motor, manufactured by the Dumore Co., Fourteenth and Racine streets, Racine, Wis., meets the require-

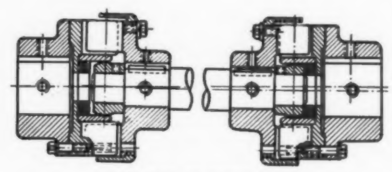
ments of the U. S. Air Corps specifications 32159 and 32160 Class "C", style IV. The motor has a rating of 1/30 horsepower at 7500 revolutions per minute, and is built for continuous duty, 12 or 24-volt operation.


HYDRON
METALLIC
BELLOWS
SERVING THE AIRCRAFT INDUSTRY

Outstanding for its compact construction, the E2X motor has an overall length of 4¾ inches and weighs 2 pounds 2 ounces. To minimize weight the case is of magnesium. The A2X type, another motor of this series, is the same as the E2X except that it is larger, developing more power, with a rating of 1/15 horsepower and a weight of 2½ pounds. The motor is explosion-proof.

Coupling Has Floating Shaft

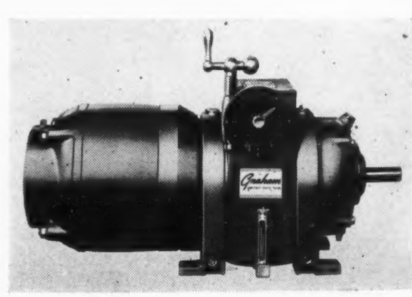
DESIGNATED as L-R Type HKQ, a new floating-shaft flexible coupling has been introduced by Lovejoy Flexible Coupling Co., 5009 West Lake street, Chicago. The coupling is applicable to either horizontal or vertical drives. Floating shaft is supported by fixed bearings, entirely independent of the



power-transmitting elements—which are unaffected by forces generated by the floating shaft, and do nothing but transmit power. Another advantage is that concentricity of ends of the fixed shafts is permanently assured, eliminating whipping of ends of the shaft at any speed. The coupling can be readily disassembled by merely reversing three to five driving bolts. This is done without disturbance of cushions or connected units. Three types of cushions are used: Metalflex, used where heavy loads develop (at speeds not exceeding 300 revolutions per minute); leather load cushions for use on sustained loads with infrequent or no cyclic variation; and multiflex cushions for fluctuating loads, the absorption of vibration and maximum misalignment. Available in standard sizes with bores from 1½ to 8½ inches (7 to 800 horsepower at 100 revolutions per minute), the couplings are suitable for various distances between fixed shafts, and for handling extreme misalignments of both the angular and parallel displacement types.

Variable Speeds—Both Directions

DEVELOPED by Graham Transmissions Inc., 2711 North Thirteenth street, Milwaukee, is a new variable speed drive unit with equal speeds forward



or reverse without stopping or reversing. The unit delivers full torque over entire speed range from maximum to zero and can hold selected speeds closely without cyclic variation as well as obtain preselected speeds accurately on reset. The reversing fea-

ture is of special value in applications to sighting, aiming and steering devices. In addition to the three sizes now available, ⅛ to 1½ horsepower, a fourth unit for 3 horsepower will soon be in production. These are offered in standard, geared reduction and geared step-up types for either built-in or coupled motor mountings.

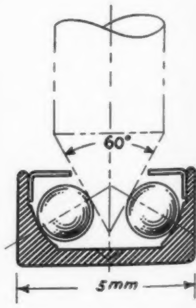
Motor Designed for Machine Tools

INTRODUCTION of a new fractional-horsepower motor, designed to meet machine tool and other industrial application requirements, has been announced by General Electric Co., Schenectady, N. Y. This motor, available in 1/4, 1/3, 1/2 and 3/4 horsepower sizes for operation on three-phase and direct

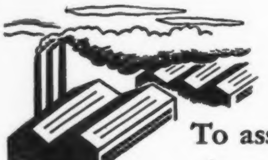
current systems, is suitable for use where frequent start-stop service, plugging and metal-dust atmospheres are encountered. Totally enclosed, the motor has a sturdy cast-iron base, closely machined endshield and stator rabbets, tough Formex wire, ball bearings, a one-piece indestructible cast-aluminum rotor, and firmly anchored windings—all providing it with a high degree of rigidity and sturdiness. Because of thrust encountered in many machine applications, ball bearing assemblies are used which allow mounting of motors with shaft at any angle to the horizontal, and rotating of the stator to any desired position. End-mounted motors can be installed vertically as well as horizontally and may be obtained for flange, flat-face, or rabbet-machine mounting.

Ball Bearing Is Self-Aligning

SUPPLEMENTING its line of miniature precision ball bearings which range in size from ⅛-inch to 5/16-inch outside diameter, Miniature Precision Bearings, associated with Split Ballbearing Corp., Lebanon, N. H., has announced a new self-aligning miniature precision pivot ball bearing. Type No. 3P is available in high-carbon chrome bearing steel (SAE 52100), machined, hardened and highly finished on raceway and exterior surfaces. The company is prepared to manufacture pivots to customer's specifications, in high-carbon chrome steel, hardened and ground to 60-degree cone. An unhardened pivot



may be used where high precision and long life are not required. Outside diameter of these antifriction



To assist manufacturers and engineers in finding practical solutions to problems involving the selection, treatment, fabrication and use of alloys containing Nickel, The International Nickel Company, Inc. have compiled essential facts based on years of research and field studies. These facts are condensed into convenient printed form.

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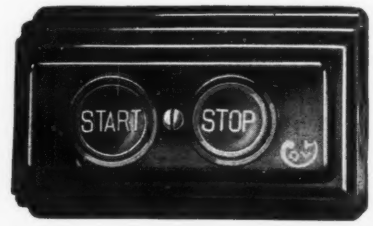
bearings is 5 millimeters; width, 2.5 millimeters; and the balls are four 1/16-inch. The minimum diameter of the pivot is about .050-inch with the maximum unlimited.

Adds Pushbutton to Line

ATTRACTIVELY styled for use with modern machines, the No. 202 pushbutton station has been added to the new line of industrial controls made by the Electrical division, Colt's Patent Fire Arms Mfg. Co. ,Hartford, Conn. The guard ring on the molded



phenolic cover is raised the full height of the start button to prevent accidental starting. Horizontal mounting is available along with the usual vertical position. Internal mechanism is self-contained on a sturdy molded block mounted directly on the metal base. Standard finish is black top with machine gray bottom, but special color combinations can be supplied.

Small Base Pump Introduced

ANOTHER new motor-driven centrifugal pump, the No. 206A, has been added recently to the line of pumps produced by Brown & Sharpe Mfg. Co., Providence, R. I. Differing only in its base, this pump is similar in design and characteristics to the No. 206 pump. To make it suitable for installation where space is limited the pump has a small base, with a depth of submergence of 9-15/16 inches. Fitted with grease-sealed ball bearings and stainless steel shaft integral with the ¼-horsepower motor, the pump is powered to handle liquids up to 1000 S.S.U. It is hydraulically balanced and has an aluminum bronze impeller which allows abrasives to be discharged without excessive wear within housing. The totally enclosed motor develops 1725 revolutions per minute. Pumping water discharge varies from 20.5 gallons per minute at 5 feet total head to 8.5 gallons per minute at 11 feet total head. When handling oil, delivery is slightly reduced.

Extruded Plastics Replace Aluminum

TO COPE with limitations in supply of aluminum due to priority demands of the Defense program, R. D. Werner Co., 380 Second avenue, New York, finishers of extruded metal molding, recently announced a new line of plastic products known as "Plastikmould" and "Plastiktrim." The two new plastic moldings are available in a wide range of

colors and a variety of shapes and sizes including rods, tubes and other commercial items both flexible and rigid.

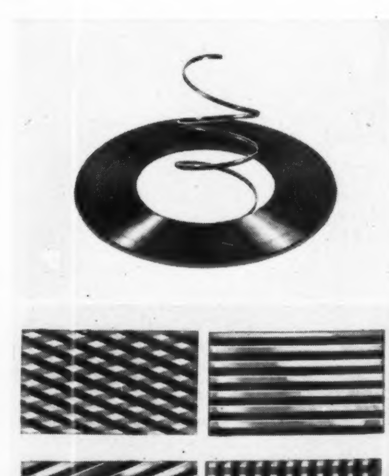
Valve Controls Piston Travel

DESIGNED and built by Hanna Engineering Works, 1765 Elston avenue, Chicago, a new two-direction valve controls the speed of pneumatic piston travel. Installed between operating valve and

one end of a cylinder, one valve provides adjustable control of inflow as well as exhaust of air independently to and from one side of piston. One valve will, therefore, control piston speed in two directions. Two control valves, one for each end of the cylinder, are recommended for extra sensitive adjustment and control of piston speed. The valve is so constructed that two adjusted orifices which control the airflow in two directions are set before flow takes place, insuring control from start of movement. Valve body is cadmium plated and all other parts are of corrosion resistant material. The valve is recommended for 250 pounds maximum air pressure, and is available in ⅛, ¼, ⅜, ½, ¾ and 1-inch sizes.

Coil and Sheet Crimp Available

SEVERAL new crimped designs in American Bonded prefinished metals have been announced by American Nickeloid Co., Peru, Ill. One of these, a

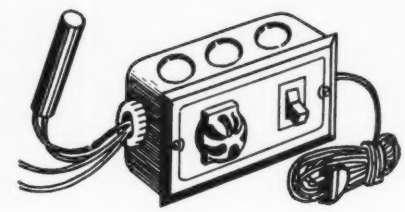


"7/16-inch crimp" is available in horizontal, diagonal, square and diamond patterns in a variety of thick-

nesses, in sheet size up to 24 x 36 inches. The second design, termed the "oval crimp" is obtainable in two widths—⅛ and 3/16-inch in long continuous coils, and in gages ranging from .010 to .015-inch. Both crimps are offered in bright or satin finishes of nickel, brass, chromium, or copper electro-bonded to basic metals of steel, zinc or brass.

Thermostat Is of Cartridge Type

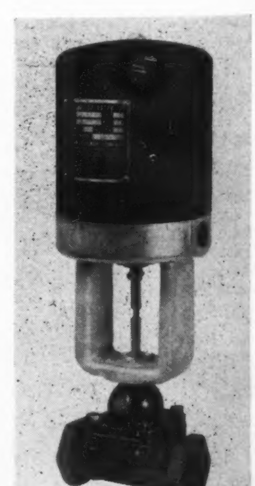
RECENTLY announced by George Ulanet Co., 89 East Kinney street, Newark, N. J., the controlling thermostat called "Pilotherm" contains a snap switch and box, pilot light and "cartridge type" thermostat completely assembled and ready for in-



stallation. The thermostat, which is ⅝-inch x 2¼ inches long, is provided with three feet of asbestos-covered nickel-stranded wire between the cartridge type thermostat and the switch box, three feet of nickel wire for connection to heater terminals and a six-foot rubber cord set at the input end of the switch box, 500 watts, 115-230 volts a.c. It is available for maximum operating temperatures of 300, 450 and 700 degrees Fahr., for use on electrically-heated machines of all types such as hot plates and tables, molding presses, etc.

Valve Adapted to Automatic Control

MOTOR-OPERATED valve particularly adapted to automatic control of corrosive fluids, liquids containing solids, viscous and volatile substances, or gases, has been introduced by Barber-Colman Co., Rockford, Ill. It may be used to control liquid level in tanks, vats, etc.; density of solutions in evaporators; flow of reagents in pH control, treatment of water, boiler feed water, etc. Valve bodies are available in various types of alloys, or cast iron lined with lead, rubber, or glass to meet requirements of a particular service. Diaphragms are tough, resilient and resistant to fluids handled. Unique construction eliminates stem packing or stuffing box and prevents corrosion, clogging or sticking of working parts. The valve is operated by a high-voltage, heavy-duty motor designed for industrial use. Wick type oilers are provided for motor bearings which are of Nitralloy. Eccentric cam and cam followers for operating valve plunger is



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FEET PER POUND

Size Awg	Heavy Formex	Enamel Single-cotton
8	19.7	19.7
9	24.9	24.9
10	31.4	31.3
11	39.5	39.4
12	50.0	49.6
13	62.9	62.5
14	79.4	78.8
15	99.0	98.9
16	125	124
17	158	156
18	199	197
19	256	247
20	315	310
21	398	389
22	503	487
23	633	611
24	794	763
25	1000	955
26	1260	1190
27	1580	1490
28	2000	1850
29	2520	2290
30	3160	2850
31	3990	3510
32	5050	4360
33	6330	5300
34	7940	6420
35	10050	7900
36	12650	9410
37	15900	11600
38	20100	14300
39	25200	16700
40	31700	19100

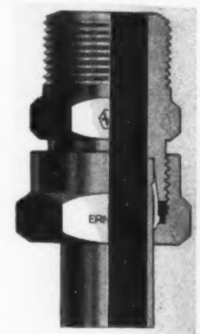
MAGNET WIRE
ALWAYS THE RIGHT TYPE

GENERAL ELECTRIC

of hardened polished steel, and packing gland around plunger prevents vapors, dust, etc., from entering through the base of the control unit. The valve is furnished in sizes of ½-inch, ¾-inch, 1 inch and 1¼-inch.

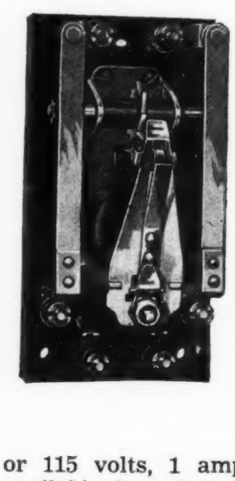
Safety Fitting for Tubing

FOR use with tubing of all metals and on hard-to-work metals such as stainless steel and aluminum, a safety fitting is now being placed on the market by The Weatherhead Co., Cleveland. The cutting edge of a hardened ring of this new fitting, known as the Ermeto safety fitting, shears a groove into outer surface of tube. The resulting leak-proof joint cannot be pulled apart and holds beyond the burst strength of the tube itself. No flaring, threading, soldering or welding is necessary, and connections made with the fitting may be taken apart easily and remade without effecting the tightness of the joint. The fittings are made of steel for high-pressure applications in aviation, machine tool, petroleum, oil refining, refrigeration, automotive, railroad, and numerous other industries. They are available as nipples, connectors, unions, elbows, tees and sleeves for tube diameters of from ⅛-inch to 1½ inch outside diameter.



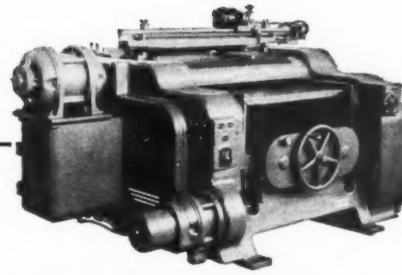
Small Ratchet Relay Available

SMALL in its physical size, the new ratchet relay announced by Struthers Dunn Inc., 1335 Cherry street, Philadelphia, offers dependable performance in opening and closing an electrical circuit over a single line. It has two independent poles and by adjustment of its cams may be made single-pole, double-break, single-throw; double-pole, single-break, single-throw; or single-pole, single-break, double-throw. Units are available for both intermittent and continuous duty. Being designed for front-connected vertical mounting, the base size is only 3-13/16 inches by 2 inches. Contact rating for noninductive load is 110 volts, 6 amperes, or 220 volts, 3 amperes alternating current; or 115 volts, 1 ampere direct current. Coils are available from 6 to 220 volts alternating current, at approximately 4 watts; or 2 to 230 volts, direct current, at approximately 2 watts. Direct current voltages above 90 requires a series resistor in the coil circuit.

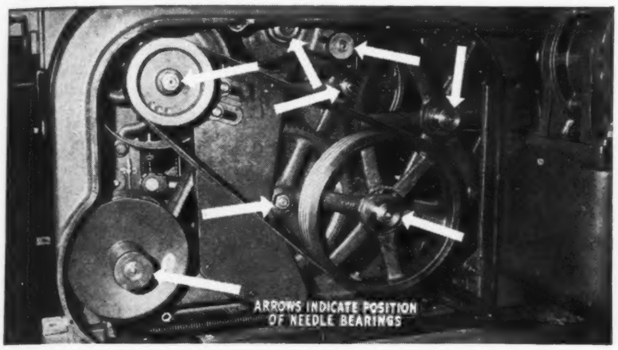


Coating Protects Surface

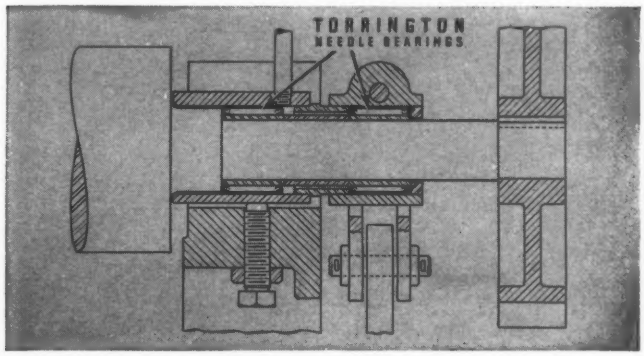
PROTECTION of metal surfaces against rust, corrosion and rot caused by salt spray, humidity, acids and other attacks is provided by Marlox, a plastic coating for metals introduced by Marley Chemical Co., 983 East Milwaukee, Detroit. Marlox



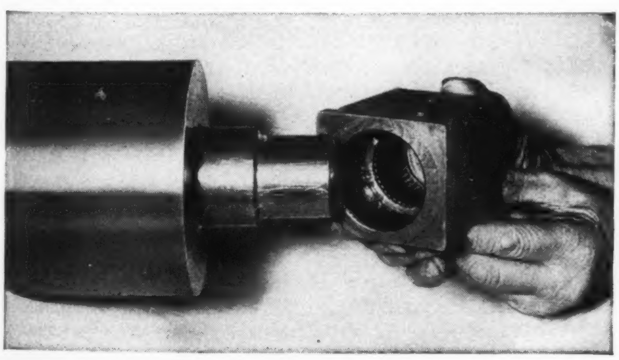
COMPACT TORRINGTON NEEDLE BEARINGS TAKE HEAVY LOADS IN BIG WHITNEY PLANERS



THIRTY NEEDLE BEARINGS are used in the new Whitney No. 97 Production Wood Planer. They carry heavy loads at the hubs of main driving gears, shown here, among many varied applications. "Other bearings would greatly increase the size of housings and necessitate staggering for clearance," reports Mr. E. D. May, Chief Engineer of Baxter D. Whitney & Son, Inc.



"BECAUSE OF THEIR UNIQUE combination of extreme compactness and high load capacity, we have selected Torrington Needle Bearings for many major applications on the big No. 97 machine," continues Mr. May. Each of the large, main gear-driven feed rolls (as shown above) operates on Needle Bearings, requiring a minimum of attention throughout their service life.



NOTE HOW READILY ADAPTABLE Torrington Needle Bearings are to this and other types of housings. Design and construction are often much simplified and bulk reduced by their use. The Needle Bearing occupies no more space than a plain bushing. It is therefore also much used in smaller assemblies.

HERE IS QUICK, EASY INSTALLATION for you! Two Needle Bearings are being assembled with pinion in an arbor press. Cost of maintenance is low, too, because lubricant is retained by the curved-in lips of the hardened retaining race—which also makes a natural seal against dust and dirt.

Have you compared the advantages of Torrington Needle Bearings with your present bearings? You will find the Needle Bearing's advantages of low coefficient of friction, very high radial load capacity, ease of installation, small size, and excellent lubrication described in great detail in Catalog No. 109. Be sure to

write for your copy. Our Engineering Department will gladly help you adapt the Torrington Needle Bearing to your special requirements. For Needle Bearings to be used in heavier service, write our associate, Bantam Bearings Corporation, South Bend, Indiana, for Booklet 103X.

THE TORRINGTON COMPANY, TORRINGTON, CONN., U. S. A. · ESTABLISHED 1866

Makers of Needle and Ball Bearings

New York Boston Philadelphia Detroit Cleveland Chicago London, England

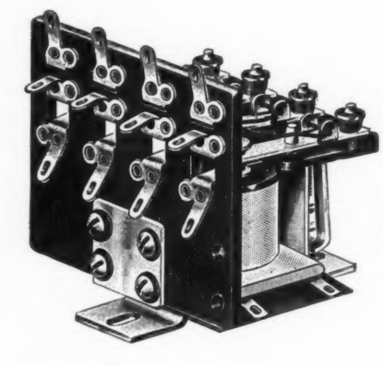
1866 75 YEARS OF MANUFACTURING PRECISION PRODUCTS 1941

TORRINGTON NEEDLE BEARING

may be applied to all types of metals and alloys including steel, cast iron, aluminum, aluminum alloys, magnesium alloys, copper, brass, cadmium plate and galvanized or zinc-coated steel. Sprayed, painted or dipped, it forms a thin, flexible coating on any metal to which it is applied—so flexible that it is not affected to any noticeable degree by temperature changes. Advantages claimed for the coating are its absence of porosity, exclusion of air film and ease of application. It can also be used where weight is a factor, as in airplanes.

Relay Added to Line

FOUR-POLE double-throw relay, newest addition to the line of type "C" relays manufactured by G-M Laboratories, 4314 North Knox avenue, Chicago, is outstanding in its precise machine assembly of parts, self-cleaning wiping action of the contacts and



long electrical and mechanical life. Under normal conditions operating voltages range from 2 to 230 volts alternating current and 2 to 125 volts direct current. Normal contact capacity is 10 amperes on noninductive alternating current loads but special contact materials for specific applications may permit control of considerably higher current. Overall dimensions of the new relay are 2 9/16 inches long, 2½ inches high, and 2⅝ inches wide for normal applications.

Motor for Dust Atmospheres

FAN-COOLED, totally enclosed motor developed by Century Electric Co., 1827 Pine street, St. Louis, has been approved by Underwriters' Laboratories Inc. for Class II, Group G atmospheres. This motor protects against explosions of grain dust in suspension. Cooling air is forced through large air passages surrounding motor closure at a velocity which resists clogging and keeps air passages clean. The motor, improved in appearance and having greater protection, is finished in smooth machine gray.

Hose Utilizes Balanced Principle

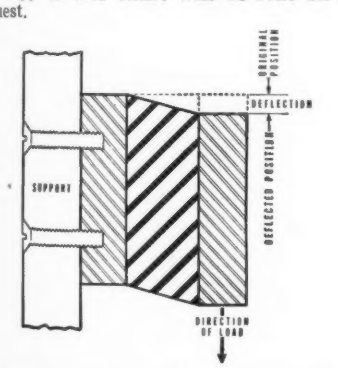
INTRODUCTION of a new type of rubber hose claimed to combine balance, homogeneity and flexibility, has been announced by the Manhattan Rubber Mfg. division, Passaic, N. J. This hose was developed after numerous experiments in the application of the Homo-Flex principle of balanced engineered construction to the hose. Among the advantages claimed for the hose are: Flexibility, lightness in weight, ease of handling, inseparable covers and plies, uniform diameters, and less elongation and expansion. It is manufactured in 50-foot lengths on accurate steel mandrels in several types covering a wide variety of service. Among

Pertinent engineering facts

About Vibration Control

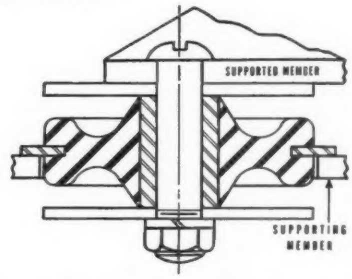
To prevent the transference of vibratory forces to or from a piece of equipment, it is necessary to approximate closely the ideal condition of a vibrating body completely free in space. A vibratory force, under such conditions is resisted only by the inertia of the body and the body will be accelerated through a small oscillatory motion. By supporting the equipment with flexible mountings this completely free condition can be simulated. The mountings, deflected by the weight of the body, accommodate the motion of the body, and only a small portion of the vibratory force is transmitted to the supporting structure.

In applying these engineering principles to flexible mounting design, it follows that for a known load and disturbing frequency, the mounting efficiency increases as its deflection increases. Increased deflection of a mounting results in lower natural frequencies of the mounted assembly. As the natural frequency decreases, the vibratory forces transmitted also decrease, as compared to the forces which would be transmitted through a solid support. This relationship has been plotted on a large four colored chart which is useful in determining the amount of vibration isolation possible to obtain in any given system. A copy of this chart will be sent on request.



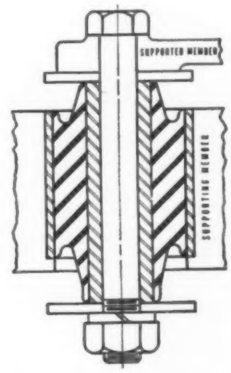
The use of a mounting in which the rubber would be stressed in shear under load, offered the one method of obtaining necessary softness in the direction of the vibratory thrust. A schematic diagram showing the principle of shear action in a mounting is illustrated at the bottom of the left column. The successful application of this type of mounting is due to the Lord method of bonding rubber to metal, which produces a bond as strong as the rubber.

Lord has perfected various styles of shear-type vibration mountings. These have been designed to function safely and efficiently at their rated loads and deflections in all types of mechanical equipment where vibration or noise exists.



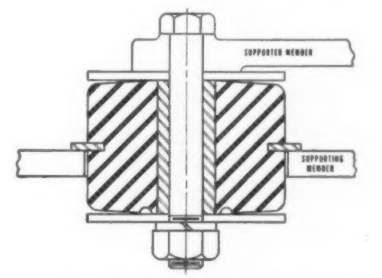
Lord Plate Form Mountings are made in numerous sizes for loads from a few ounces to 300 pounds. The diagram above shows their construction and typical method of installation. In this mounting the end shape design prevents stress concentration at the bond. Snubbing shoulders arrest shock movement and supply a cushioned stop on contact with metal washers. Plate Form Mountings have approximately twice the resistance to radial movement as they have to axial movement.

Tube Form Mountings are light weight, compact units designed for loads from a few pounds to 1500 pounds. In radial direction these mountings have much greater resistance to movement than in the axial direction. This design is based on the use of a safe stress concentration of 40 pounds per square inch on the projected area of the center pin. In this design also, the end shape distributes the load evenly through the rubber body. Tube Form Mountings are readily installed by pressing or clamping them into a socket. In addition to their function of isolating noise and vibration, these mountings often eliminate ex-



tremely accurate machining operations. The bonded rubber construction will compensate for misalignment in the installation without affecting the efficiency of the mounting system.

Many types of equipment set up shock loads in addition to normal vibratory forces. To efficiently isolate these shock loads, Vertical Snubbing Mountings are available in both plate and tube forms. The large snubbing area on these mounts is evident in the V. S. plate form installation drawing below.



The Lord mountings described on this page are being used successfully in practically every industry. The vibration engineering experience of Lord engineers gained in working vibration problems is available without obligation. A letter, outlining the design of your equipment and giving the details of your problem will enable us to make recommendations.

LORD MANUFACTURING COMPANY . . . ERIE, PA.

923 FAIRMOUNT ROAD, BURBANK, CAL. 280 MADISON AVE., NEW YORK 332 S. MICHIGAN AVE., CHICAGO

IT TAKES RUBBER IN SHEAR TO ABSORB VIBRATION

these are air hose, water hose, high pressure orchard spray hose, mine spraying hose, air-oil and oil spray hose, etc.

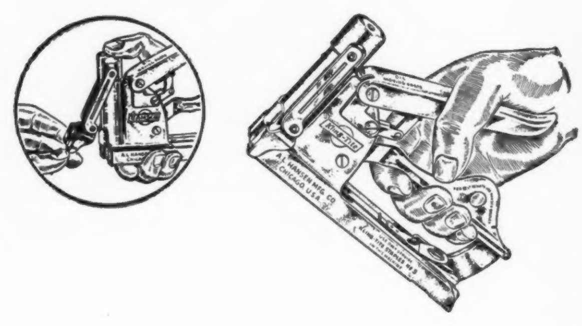
Pressure Switch Developed

PRESSURE switch No. 611-030 annouced by Jefferson Electric Co., Bellwood, Ill., is designed for use on water pump, pneumatic and oil systems, to control stopping and starting of small single or polyphase alternating current motors on predetermined pressure settings. The switch has a working pressure range up to 80 pounds and the differential between "on" and "off" positions may be specified and adjusted according to requirements. Differential is usually 20 pounds.

Engineering Dept. Equipment

One-Hand Tackers Improved

ONE-HAND tackers produced by A. L. Hansen Mfg. Co., 5009 Ravenswood avenue, Chicago, have been improved in design, and utilize a new take-up jaw which permits tacker to be opened at front for convenient and quick inspection. In a few sec-



onds the jaw can be replaced for continued tacker operation. It is removed by pressing down on cap and swinging jaw down and outward. Reversing the procedure puts the jaw back in place. This tacker is one of a line of thirty-six models made by the company.

Sensitized Paper Improved

SENSITIZED blue line paper for use with white print dry developer or ammonia vapor machines has been improved by Frederick Post Co., Hamlin and Avondale avenues, Chicago. The improvements are 50 per cent rag content bond; two speeds in sensitivity—regular and fast; and two colors—deep royal blue and "Post" red. The new sensitizing medium prints out to a cleaner, whiter background and at same time leaves all lines in deeply colored contrast. Special emphasis is placed on fine lines.

Pencil Is Hard and Strong

SIX graded degrees for every requirement are available in the almost unbreakable pencil offered by Reliance Pencil Corp., Mount Vernon, N. Y. Because of its strength it is excellent for office use or drawing. It is also good for carbon-copying as lead gradation is soft enough to make legible originals, yet strong enough to hold up under pressure necessary to make multiple copies.

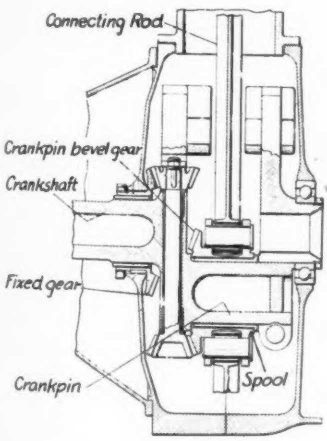
Noteworthy PATENTS

Effects Uniform Radial Piston Motion

CONVENTIONALLY, two methods are available for attaching connecting rods of radial engines to the crankshaft. In the first of these, one of the connecting rods is made a "master" rod and all other rods are connected to it. This results in differences in motion of the several pistons.

In the second method, the rods are attached to a spool, mounted on the crankpin, which is constrained so that rotation about its own axis is impossible. Assigned to the Lawrance Engineering and Research Corp., the patent discussed herein provides a means for improving the rotational constraint of this spool by the use of bevel gears.

Two bevel gears are used, one integral with the spool on the crankpin, the other concentric with



Bevel gears and pinions maintain precise orientation of the crankpin spool in relation to the pistons in a radial bank

the crankshaft and integral with the housing. An intermediate shaft, bushed in the crankarm, carries a bevel pinion on each end. Each pinion meshes with one of the two bevel gears. Thus, so long as the ratio between each mating gear and pinion is the same, the spool will be restrained from rotation about its own axis and each piston will have a motion identical with all others.

In the case of certain numbers of cylinders in a radial bank, reversals of load are experienced which tend to cause the spool to oscillate on the crankpin. In order to avoid consequent shock loading in the gearing it may be desirable to provide some form of resilient anchoring for the fixed bevel gear. This may be accomplished by attaching a radial arm to the fixed gear and cushioning the end of

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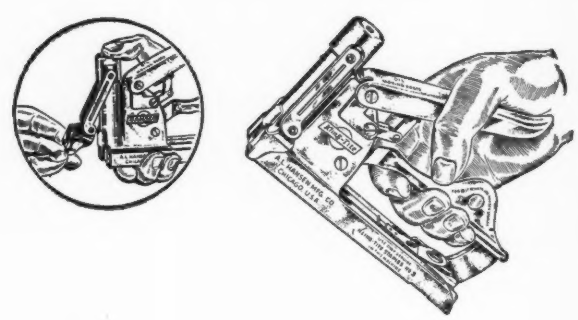

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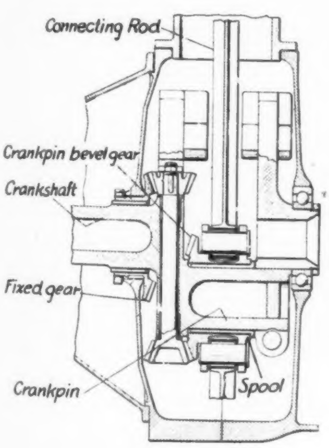


Bevel gears and pinions maintain precise orientation of the crankpin spool in relation to the pistons in a radial bank

In the case of certain numbers of cylinders in a radial bank, reversals of load are experienced which tend to cause the spool to oscillate on the crankpin. In order to avoid consequent shock loading in the gearing it may be desirable to provide some form of resilient anchoring for the fixed bevel gear. This may be accomplished by attaching a radial arm to the fixed gear and cushioning the end of

this arm between springs or other shock absorbing members.

Performs Double Timing Operation

POSITIVE controlled time delay between the opening of a control switch and the closing of the relay contacts as well as between the closing of the control switch and the opening of the relay contacts is provided in this patent assigned to the Adlake Company.

Utilizing mercury as a contact medium the complete relay is enclosed in a non-conducting container surrounded by a field coil. Instead of ordinary orifice apertures, porous ceramic plugs are used. These may be of alundum or equivalent gas-porous material. Such plugs have the advantage over orifices in that they are impervious to the passage of mercury.

An operating cylinder closed by a plug on one end and open on the other is fixed within an armature capable of longitudinal movement within the outer container. To provide delay before opening the contacts a contact hood is provided which covers the contact inside the operating cylinder.

In the position shown on the left the switch has just been opened, permitting the armature (and operating cylinder) to fall into the mercury reservoir. As the mercury level rises, the gas used as the timing medium flows through first the upper plug and then through both plugs until the circuit contact is completed.

If the switch is then closed, the armature is raised

(*Concluded on Page* 112)

Fully enclosed mercury time delay relay utilizes a porous ceramic material which is impervious to the passage of mercury

(*Concluded from Page* 106)

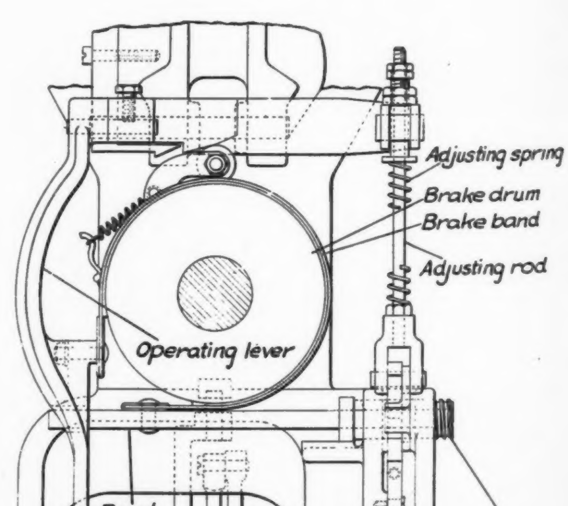
but mercury is trapped in the contact hood as shown on the right. Gas then flows through the lower plug as the mercury level falls, until the contact is broken.

Compensates for Brake Band Wear

PRESSES of the crank or eccentric type are usually fitted with a band brake which is applied automatically at the end of each press cycle. In the illustrated mechanism this is accomplished by a downward movement of the operating lever which wraps the band tightly around the drum. Since the length of stroke of this lever is constant, wear of the brake band will ultimately result in a failure to stop the press. Assigned to the United Shoe Machinery Corp., this patent discloses means of pretensioning the band at each cycle of press operation, thereby compensating for wear.

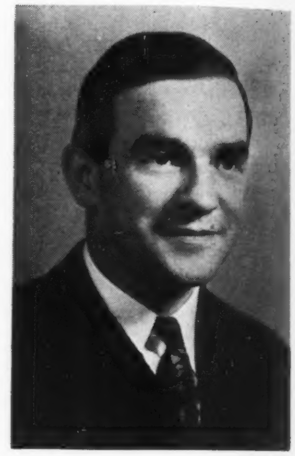
Instead of anchoring one end of the brake band rigidly to the frame, it is fastened to a bar or band anchor which may be moved longitudinally by rotation of a nut on the adjusting screw. Affixed to this nut is a ratchet wheel. A crank, which carries a pawl engaging the ratchet, is attached to the operating mechanism by means of an adjusting rod. The opposite end of this rod floats in a hole in the operating lever. An adjusting spring maintains the operating lever in contact with stop nuts.

During each cycle of operation of the press (about 90 degrees before the brake is applied) the operating lever compresses the adjusting spring. If the tension in the brake band is inadequate the spring transmits the motion to the rod which, through the ratchet and nut, moves the band anchor to the left and adjusts the band.



Automatic take-up of press brake band prevents over-cycling resulting from wear

MEN Of Machines



FORMERLY manager of the Westinghouse gearing division, R. S. Marthens has been appointed staff assistant to the manager of the Canton Ordnance division, which when completed will be operated by Westinghouse and owned by the U. S. Government. A native of Conneaut, Pa., Mr. Marthens joined the Westinghouse company as a youth and studied engineering at its night school. From 1917, with the exception of service with the U. S. Army air corps during 1918-1919, he was a member of various engineering divisions of the company. He was first assigned to the power engineering department where he remained until 1930, when he was transferred to the gearing division as section engineer. In 1932 he was made manager of engineering for the division and two years later was appointed gearing division manager, holding this position until his recent appointment.

RECENT appointment as chief engineer of Acme Steel Co., Chicago, gives due recognition to R. E. Orton and to his experience in the field of design. After graduating from the Armour institute he joined the Orton Crane and Shovel Co. in 1928, and for the last three years of his connection with this company was chief engineer. In 1936 he became associated with Wisconsin Steel Works as special investigating engineer and a year later

joined Acme Steel Co. His experience includes that of being in complete charge of the design and development of a variety of special machines such as locomotive cranes and shovels, both large and small. He also obtained considerable experience in motor and special control applications. Having studied the proper application of metals and their heat treatment from the theoretical as well as the practical side, he is well qualified to fill his new position. Mr. Orton is well known to MACHINE DESIGN's readers through his outstanding contributions to this journal.

♦

H. OLIVER WEST has been elected executive vice president, Boeing Airplane Co., Seattle, and its subsidiary, Boeing Aircraft Co. Mr. West started with Boeing in 1921, and became assistant to the president in 1939.

♦

WILLIAM L. BATT, president, SKF Industries Inc., has been awarded the 1940 Henry Laurence Gantt memorial gold medal for "distinguished and liberal-minded leadership in the art, science and philosophy of industrial management in both private and public affairs." This award is presented annually by the American Society of Mechanical Engineers and the Institute of Management.

♦

FRED W. BUSH, who has been connected with transformer engineering, development and sales since 1930, has been promoted to engineer-in-charge of transformer sales of Allis-Chalmers Mfg. Co. W. C. SEALEY has been appointed engineer in charge of transformer design. He has been connected with the transformer division since 1931.

♦

CHARLES H. COLVIN has been appointed director of the Daniel Guggenheim School of Aeronautics, New York university. He has been acting chief of the instruments division, U. S. Weather Bureau, Washington. Mr. Colvin will succeed DR. ALEXANDER KLEMIN who is retiring in September.

♦

FRANK J. ZINK, formerly professor of agricultural engineering, Kansas State College, and more recently associated with farm equipment companies in a research capacity, has been appointed to the headquarters' staff of Farm Equipment institute, Chicago.

♦

GLEN T. LAMPTON has been appointed assistant engineer in charge of experimental engineering, Hamilton Standard Propellers division, United Air-

craft Corp., East Hartford, Conn. He was formerly engineering manager, Lycoming division, Aviation Mfg. Corp. Carl F. Baker, assistant engineer, will head the production engineering section, and Thomas B. Rhines, assistant engineer, will also assist. The development engineering section of the company will be in charge of Murray C. Beebe, assistant engineer, and Charles B. Conwell will be project engineer.

♦

Forest S. Baster, with White Motor Co. since 1936, has been promoted to vice president in charge of engineering in recognition of his work in designing the company's line of super-power trucks. Robert Cass, formerly executive engineer, has been named chief engineer; and O. F. Quartullo, special design engineer, has been appointed chief engineer of the White Horse division.

♦

L. E. Osborne has been granted leave of absence from his duties as manager of manufacturing and engineering of Westinghouse Electric & Mfg. Co., to join the staff of the Defense Commission's office of Production Management in Washington. John H. Ashbaugh, formerly engineering manager at East Springfield works, has been appointed to succeed Mr. Osborne.

♦

W. K. Ebel, chief engineer of Glenn L. Martin Co., has been made vice president in charge of engineering and director of the company. He has been responsible for engineering as well as flight testing of all the modern bombers manufactured by the company for the Army, Navy, and British. He will retain his old title of chief test pilot.

♦

H. H. Beverage has been made vice president in charge of research and development of RCA Communications Inc. Since 1932 he has been chief research engineer of the company.

♦

Frank W. Curtis, chief engineer of Van Norman Machine Tool Co., and chairman of the Springfield, Mass., chapter of the American Society of Tool Engineers, has been elected new president of A.S.T.M.

♦

Harry I. Strube, formerly assistant chief engineer at Philadelphia for Link-Belt Co., Chicago, has been appointed chief engineer of the Eastern division of the company with headquarters at the company's Philadelphia plant. He will succeed F. F. Waechter, resigned after 43 years' service.

♦

E. C. Hach of Westfield, N. J., has joined the staff of the research laboratory of American Society of Heating and Ventilating Engineers. Mr. Hach's previous experience includes that of chief engineer of the manufacturing department of Standard Air Conditioning Inc., a division of American Radiator & Sanitary Corp., New York, where he was in charge of research development and design. Prior to this connection he was employed by the Carrier Corp. on research and development.

Streamline MACHINERY INSTALLATIONS

with UNISORB* Foundation Felt

WHY put time and labor into bolting production machinery to the floor when 90% of it does not need it? Unisorb Foundation Felts and cement save time and money over the old-fashioned way. Experience has proved, also, that Unisorb Foundation Felt reduces noise and isolates vibration, thereby in many cases giving increased production from the same machine.

•

Write today on your company letterhead for our illustrated booklet, *"Unisorb in the Industrial Plant."*

*Reg. U. S. Pat. Off.

THE FELTERS COMPANY, INC.

210-E SOUTH STREET, BOSTON, MASSACHUSETTS

OFFICES: New York, Philadelphia, Chicago, Detroit SALES REPRESENTATIVES: Dallas, Los Angeles, Nashville, St. Louis, St. Paul

MILLS: Johnson City, New York • Millbury, Massachusetts • Jackson, Michigan

FELTERS FELT FUNCTIONS

Automatic Press Meets Current Conditions

(*Concluded from Page* 48)

period the timing motor is automatically stopped, and only after the curing period has elapsed does the timing motor resume operation. Again, the open contactor circuit is closed and the bottom ram moves downward until it reaches a limit switch. The motor incorporates a built-in, spring opposed, magnetic brake which assures almost instant stopping.

The controller next energizes a solenoid air valve which releases one hundred pounds air pressure through copper tubes to blow the completed molding from the mold and clean the latter for the next operation. This completed piece lands on a sensitive balance and, if the piece is complete and perfect, it will tip the balance and close a circuit. Should the piece weigh too little, it is indicative of an imperfect molding and that a part of it might still remain in the mold, in which case the circuit is such that the main circuit breaker opens and de-energizes everything except an alarm bell which indicates that something is amiss.

Safeguards Against Overtravel

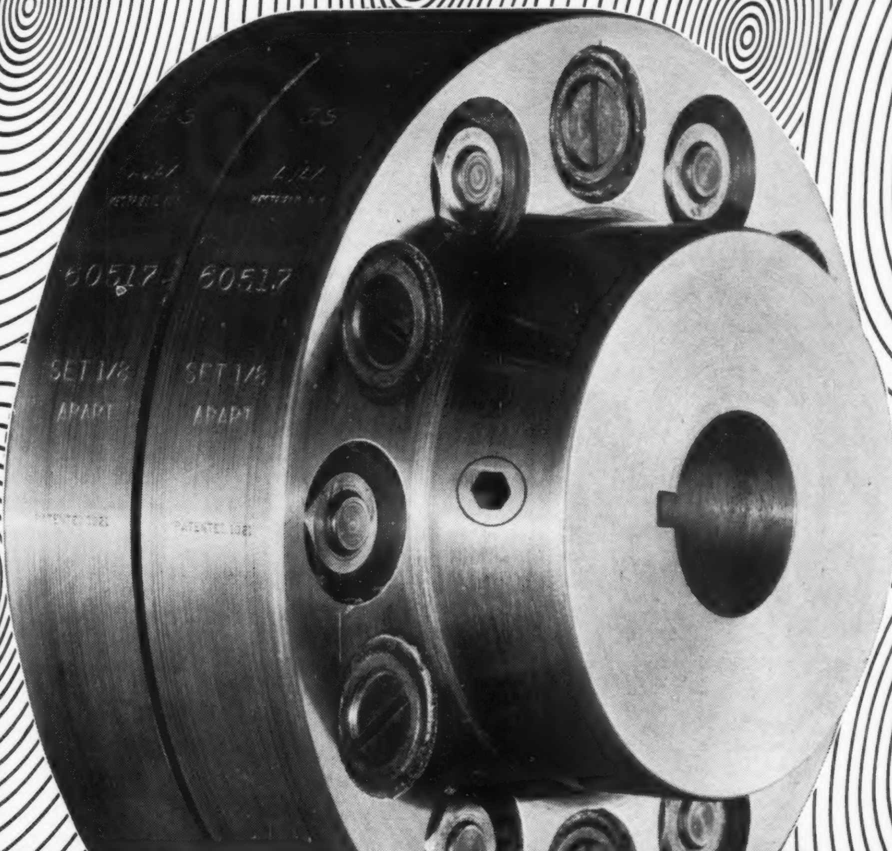
In addition to the normal limit switches, extreme limit switches are used which stop the machine should it overtravel because of some electrical defect. When the overtraveling occurs and the main circuit breaker opens, a normally closed interlock circuit is re-established which will permit the operator to move the platens from the extreme position to a normal position by merely pressing the lower button of the control shown in *Fig.* 7.

The "close" and "open" buttons are jog buttons which operate only when the main switch is in the left position. In placing molds it is desirable to jog the platens up or down in small degrees for rapid installation. When this control is directly forward the entire machine is inoperative and, when to the right, all circuits are established to produce moldings.

Cartridge Heaters Effect Curing

The molds are heated by electric cartridge heaters and their temperature is controlled by the automatic thermostat controls shown in the lower right of *Fig.* 7, one for the upper and the other for the lower press platens.

An extra safety feature is incorporated in the controller circuit in which a main controller is used in addition to the reversing controllers. This main controller is complete with overload thermal relays as is customary on installations such as elevators in which an extra degree of safety is desirable. Should the contacts of a reversing contactor "freeze," this safety feature assures that the main controller will open.



Professional Viewpoints

(*Concluded from Page* 79)

alternating current is superior from a weight saving standpoint to three-phase, 400-cycle alternating current. Single-phase equipment requires capacitor-start motors. Three-phase equipment does not require capacitors for starting and has utilization factors most nearly approaching direct current equipment

8. On all alternating-current equipment, care must be taken to maintain as high a value of power factor and efficiency as practicable from a weight standpoint
9. Considerable design study must be given toward a balance between maximum motor speed and the weight of the gear reduction required to reduce the r.p.m. to a useful value
10. Alternating current can be "switched" more easily than direct current, especially at 120 volts or more.

Possible disadvantages incurred with the use of alternating current may be briefly summarized as follows:

1. At present, alternating-current motor designs, both single phase and three phase have tended toward relatively low efficiencies and poor power factors
2. On applications using small, high-speed alternating-current motors and large gear reductions, difficulty is expected in providing enough starting torque to overcome viscous lubricant in reduction gear at low starting temperatures
3. Considerable difficulty in properly lubricating bearings and gears on high speed motors is anticipated. The importance of the bearing lubrication problem can be realized when motor armature speeds are considered as high as 24,000 to 48,000 r.p.m., depending on the number of poles and the frequency of the supply system
4. There is considerable difficulty in paralleling single-phase alternators; this difficulty, however, is greatly reduced on three-phase machines.

—E. E. MINOR, *Electrical Engineer*
The Glenn L. Martin Co.

". . . engineering inspectors needed"

To the Editor:

There is an urgent need for qualified persons to fill positions as inspectors of aeronautical engineering materials in the Navy Department. The Civil Service Commission has recently announced an examination to fill these important national defense positions.

Applications for examination should be made to the Commission's Washington, D. C. office. Forms may be obtained from any first or second-class post office.

—WM. C. HULL, *Executive Assistant*
U. S. Civil Service Commission

Selecting Special Motors

(*Continued from Page* 65)

The watts loss W will therefore be

$$W = \frac{J_{\text{(total)}}}{t_e} \text{ watts} \quad \ldots\ldots (11)$$

where t_e is the effective cooling time in seconds and is the sum of all the component intervals of time of one reversal when each is reduced to the equivalent cooling time at full speed. Its evaluation is one of great importance.

From kinetics it is known that the time required to accelerate a rotating mass to a definite speed with an average torque of $T_{(av)}$ is given by the expression

$$t = \frac{WR^2 \times N}{308 \times T_{(av)}} \text{ seconds} \quad \ldots\ldots (12)$$

where t = time in seconds, N = speed in r.p.m. and $T_{(av)}$ = average torque in lb.-ft.

It must be noted that $T_{(av)}$ is the average torque in time and cannot be obtained as the average under the speed-torque curve. Stated in words: The average torque can be found as the reciprocal of the average of the sum of the reciprocals of a convenient number of torque readings taken at equal intervals from the speed-torque curve. In general form

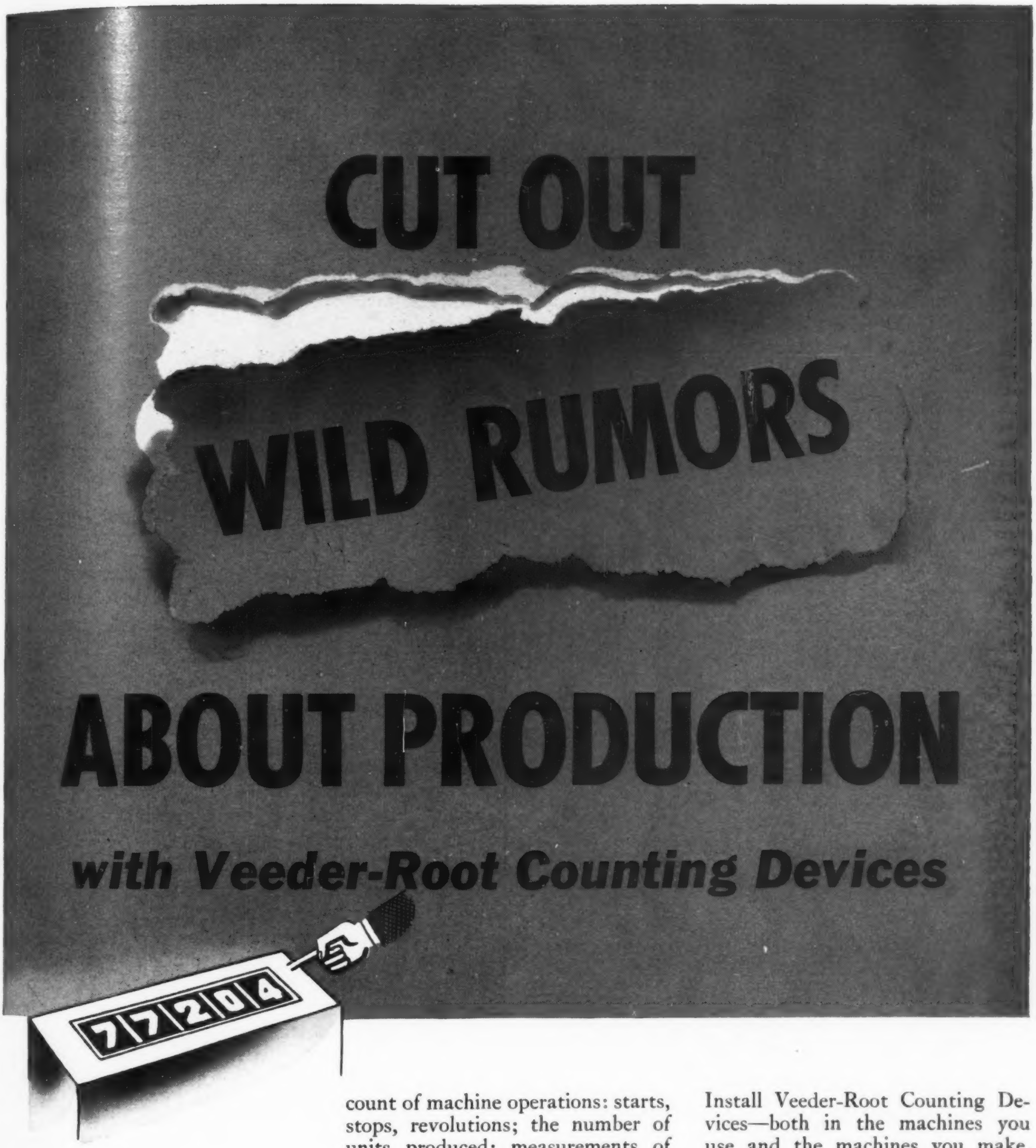
$$T_{(av)} = \frac{S_i - S_f}{\sum_{S_i}^{S_f} \frac{\Delta S}{T}} \quad \ldots\ldots (13)$$

where ΔS = interval of slip between torque readings, S_i, S_f = slip (initial and final), and T = torque in lb.-ft.

It will perhaps be found most convenient to take ΔS as .1 and to read 10 values of torque.

In conjunction with the subject of accelerating and plugging time, *Fig.* 3 is an oscillogram of current taken on a 405 frame motor with the following specifications: 20 h.p., 900 r.p.m., 400 volts, 3 phase, 60 cycles. The rotor, showing lightweight construction, and with vent holes and long bars for more effective heat dissipation, has a WR^2 of 13.7 lb.-ft.^{2} as determined by test on moment of inertia rails. The oscillogram shows the following portion of its cycle: Accelerate from rest to full speed, run in forward direction, plug to standstill, and accelerate to full speed in reverse direction. The speed-torque curve, as determined by a dynamometer, shows an average accelerating torque of 243 lb.-ft. and an average plugging torque of 311 lb.-ft. The time-torque curve as calculated from the speed-torque curve is superimposed on the oscillogram for convenience sake.

Two familiar types of torque curves are shown



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in *Fig.* 4, that belonging to the group of normal-starting torque, normal-starting current (N.E.M.A. class A) and that of high-starting torque, high-slip, low-starting current (N.E.M.A. Class *D* squirrel cage motors). TABLE V shows that in the accelerating region the Class *A* torque curve has a higher average value than the Class *D* although in the plugging region its average is about one-half the latter. When the average torque is taken over the whole range, Class *D* motors have a 30-40 per cent greater value than Class *A*. This characteristic makes itself felt directly in the total time required for acceleration and plugging as seen in *Fig.* 4.

In determining t from Equation 12, it is assumed that the only resistance to the change of speed is inertia. In many cases of automatic machine operations, load torque is applied during the accelerating period. This has the effect of lengthening the accelerating time. In general

$$\text{Effective torque} = \text{motor torque} \pm (\text{friction} + \text{load torque}) \quad \ldots\ldots (14)$$

The sign will be minus during acceleration and plus during deceleration.

It is now possible to calculate the effective cooling time. Since there is heat radiation even at standstill t_e can be taken as

$$t_e = \Sigma\, C t_{(acc)} + C t_{(dec)} + t_{(run)} \text{ seconds} \quad \ldots\ldots (15)$$

where $t_{(acc)}$ = accelerating time in seconds, $t_{(dec)}$ = decelerating time, $t_{(run)}$ = full speed running time, and C is a coefficient depending on the motor class.

The coefficient .6 can be used for C for a motor with N.E.M.A. Class *A* torques or .7 for one with Class *D* torques. The reason for this difference can be easily seen from a study of TABLE V. Note that in addition to having a higher average value of torque, Class *D* motors are rotating from half to full speed for some 60 to 65 per cent of the accelerating and plugging time and thus gain by the correspondingly better ventilation. This percentage may be contrasted with the 40 to 50 per cent for Class *A* motors. For a given number of reversals a minute it is to the designer's advantage to make $t_{(acc)}$ and $t_{(dec)}$ as short as possible so as to permit longer running time.

It has been shown previously to what devices the engineer can resort in reducing rotor accelerating

TABLE V

Comparison of N.E.M.A. Classes A and D Motors

Conditions	N.E.M.A. Class A	N.E.M.A. Class D
Av. accelerating torque (% full load)	209	180
Av. plugging torque (% full load)	146	295
Av. torque, accel. & plug. (% full load)	171	225
Time to accelerate to half speed (seconds)	2.0	1.2
Time to accelerate from half to full speed (seconds)	1.4	2.7
Total accelerating time (seconds)	3.4	3.9
Total plugging time (seconds)	4.9	2.5
Total time, accel. & plug. (seconds)	8.3	6.4
Time motor is running between half and full speed (seconds)	3.8	4.0
Total time motor is running between half and full speed (per cent)	46	62

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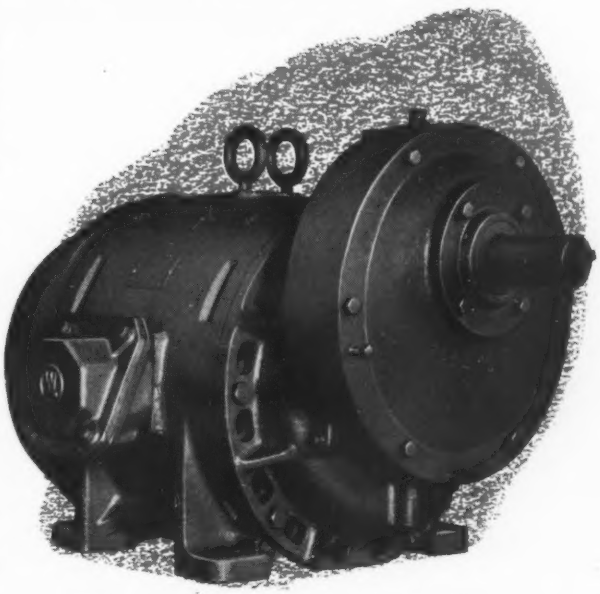

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joules. The discussion has now been carried to the point where it can be shown how the stator accelerating loss can be minimized. By referring to Equation 10 it is seen that this consists mainly of a reduction in the ratio of r_1/r_2. By special constructions the stator can be loaded with copper, thus keeping the resistance r_1 a minimum. In addition the use of a high resistance rotor can aid, even more, the reduction of r_1/r_2. There is, however, a limit to the indefinite increase of r_2. Up to a certain point, as r_2 increases the average torque increases and the accelerating and decelerating time decreases. Beyond this point there is a decrease in torque, shown by the dotted curve in *Fig.* 4. Likewise, a value of r_2 which is higher than good design dictates will result in disproportionately high running losses and will contribute more heating than is eliminated by the resulting small value of r_1/r_2. Numerous examples can be cited to illustrate the increase in permissible number of reversals with an increase in rotor resistance but the typical comparisons shown in TABLE VI should suffice. Values shown are for a 254 frame, 5-horsepower, 1800 r.p.m. open motor with Class *A* insulation.

TABLE VI

Permissible Reversals with Reference to Rotor Resistance

Type of Motor	Remarks	Reversals for 50°C. Rise (Rotor free)
N.E.M.A. CLASS A	Standard characteristics $WR^2 = .80$	16
N.E.M.A. CLASS D	Same stator as above but with high resistance rotor	24
N.E.M.A. CLASS D	Special stator, special rotor	33

Enclosures on rapid-reversing motors are of extreme importance. By far the vast majority of such applications have required open construction only but in the past several years there has been an increasing demand for more protective enclosures, and even totally enclosed fan-cooled reversing motors are now in frequent use. An excellent example is illustrated in *Fig.* 1, where fan-cooled reversing motors drive the spindles and index of a high-production automatic machine.

There are dangers and some exceptions inherent in any generalization and with this in mind it can be said that approximately one-half as many reversals can be obtained from a totally enclosed fan-cooled motor as from the identical motor when built in an open construction. The reasons for this are not hard to find. In the analysis of Equations 10 and 11 we were not critical of the extent to which each of the component losses contributed to stator heating.

In an open motor with high starting torque characteristics a large part of the total losses are generated in the rotor itself. Thanks to the mechanical construction of an open frame, most of the rotor heat is transferred by convection directly to the outside and only a small percentage finds its way

across the gap to the stator winding. Especially is this true with long bar rotor construction. Reference to *Fig.* 6 shows that the percentage of rotor loss which contributes to stator heating can be determined for either short or long bar rotors. Thus for an open motor, when calculating the watts effecting stator temperature rise, Equations 10 and 11 must be modified to the extent of including only that percentage of the accelerating and full-load running joules of the rotor as given in *Fig.* 6. The thermal dissipation of an open frame is a function of its active material (D^2L), peripheral velocity and particular mechanical construction, and hence varies between competitive designs. Knowing, from past tests, the effective watts that cause a definite stator temperature rise in a given frame, the engineer can calculate the permissible reversals per minute for that same temperature rise.

Today motors are used extensively for individual drive for quick starting and stopping motion on a large number of metal-working machines including drilling and tapping spindles, chucking lathes, broaches, special purpose production lathes, indexing drives, and many others. In other fields there are also numerous applications such as paper winders, automatic production paint sprayers, plastic presses, and other special process machines. Many of these machines have high inertia rotating parts, therefore the design of the motor for each may vary considerably depending upon the purpose for which it is used.

Success Depends on Control

Much of the success of a reversing motor application is dependent upon the accuracy of the control equipment, particularly in connection with fast operating cycles where the motion of the driven parts must be held within close limits. Where the cycles occur at the rate of thirty per minute or more the reversing contactors should be positive in action and have low inertia, so as to eliminate lost motion. These same requirements would also apply to auxiliary control such as limit switches and pushbuttons.

On a fast operating cycle the ordinary current overload protective device which operates on the inverse time element principle should not be recommended as the sole means of protecting the motor. On many of these applications the average current taken by the motor may reach a relatively high value due to the frequent starting and stopping, and a current overload selected for these conditions would offer comparatively little protection to the motor. It has been found that the thermal element which is applied directly in contact with the stator core of the motor will afford protection against operating the motor beyond its capacity. This type of element functions at a predetermined temperature of the motor core and offers a positive protection against operation of the motor in excess of its capacity. Inasmuch as this type of thermal element does not afford protection against short circuits, grounds or stalled condition a current overload in the starter should be used in addition.

Theory of Elasticity in Practical Design

(*Continued from Page* 75)

stresses from which apply until the next loaded section is reached. The loads on this section are then combined with those on the first to give a new P, e, and M, the stress system resulting therefrom then applies until the next loaded section is reached. This treatment is that which is followed in the case of a straight beam under a multitude of concentrated loads.

Of course, as always in such cases, the stress system does not apply in the vicinity of the points of application of the loads. As before, such points must be treated as local problems and then combined with the main system.

The above can best be made clear by a number of specific examples. As the simplest case consider the link of Fig. 40. Here $k = 1.125/1.875 = .6$, $r_o = (1.125 + 1.875)/2 = 1.5$, $d = .75$ and $e = .86$. From *Fig.* 39, $K_1 = 1.20$, $K_2 = .855$, $K_3 = 1.30$ and $K_4 = .78$. The stress at r_1 is, from Equation 103,

$$S_\theta = \frac{6}{.3125 \times .75^2}\left[1.2\,(-2166 \times .86) + 1.3\,(2166 \times 1.5 \sin\theta)\right]$$

This is a maximum at $\theta = 90$ degrees and equals $+67700$ pounds per square inch. The stress at r_2 is similarly obtained from Equation 104. Its maximum value is at $\theta = 90$ degrees also, and equals —32100 pounds per square inch. *Fig.* 41 is a photoelastic picture of this same link. $H = 87.2$, model load $= 82$ pounds, 8¾ fringes at r_1 and 4¼ at r_2. The stress experimentally determined is obtained from the following:

$$\frac{8.75 \times 87.2 \times 2166}{.3125 \times 82} = 64{,}500 \text{ pounds per square inch}$$

This is 4¾ per cent below the theoretical figure, which is within the experimental limits. Similarly at r_2 the picture gives 31,300 pounds per square inch, 2½ per cent low.

Figured as an eccentrically loaded straight column, the stress is given by $P/A \pm Mc/I$, which gives at r_1, $+56350$ and at r_2 —38050. In the first case the true stress is 20 per cent higher, in the second, 16 per cent lower.

In *Fig.* 42 is shown a curved rocker link pivoted on the pin at the force R, with 2000 pounds applied vertically to the right end pin, and Q at the angle shown at the left pin. Triangulation of forces determines Q, and R with its direction. $k = 4/8 = .5$ and from *Fig.* 39, $K_1 = 1.29$, $K_2 = .82$, $K_3 = 1.43$, $K_4 = .715$, and $6/bd^2 = 6/(.5 \times 4^2) = .75$.

To analyze the portion to the right of R the link

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is rotated 90 degrees as shown in *Fig.* 43*A*, the portion shown in solid lines being that under consideration. $P = -2000$ pounds and $e = -6$ inches for this arrangement. From Equation 103 the stress at r_1 is

$$S_\theta = .75\,[-1.29\,(-2000)\,(-6) + 1.43(-2000 \times 6)\sin\theta]$$
$$= -9000\,(1.29 + 1.43\sin\theta)$$

applying between $\theta = -90$ degrees and -30 degrees 45 minutes. Substituting gives for these two limits, $+1260$ pounds per square inch and -5000. It is apparent from the equation that intermediate values of θ will give stress values between these limits. From Equation 104 the stress at r_2 is

$$S_o = -.75\,[-.82\,(-2000)\,(-6) + .715\,(-2000 \times 6)\sin\theta]$$
$$= 9000\,(.82 + .715\sin\theta)$$

giving at the two limits of θ, $+950$ and $+4075$.

To analyze the left portion of the link the 2000 pound force is combined with R. This resultant will, of course, balance Q. The link is now rotated 40 degrees from its position in *Fig.* 42, as shown in *Fig.* 43*B*. P is now equal to $+3110$ pounds and $e = +3.86$ inches. Substituting in Equation 103 gives at r_1

$$S_\theta = .75\,(-1.29 \times 3110 \times 3.86 + 1.43 \times 3110 \times 6\sin\theta)$$
$$= 2330\,(-4.97 + 8.58\sin\theta)$$

and the extremes are, $\theta = 19$ degrees 15 minutes, -5000; $\theta = 90$ degrees, $+8420$; and $\theta = 140$ degrees, $+1260$. Similarly, by substituting in Equation 104, the stress at these three values of θ are $+4075$, -2610, $+950$, respectively.

The stress values at these points are shown on *Fig.* 42. Inspection of the equations shows that the stresses at intermediate points will lie between these limits. The values at the three load points must be recognized as theoretical figures only, the actual figures depending upon the method of application of the loads. At the two end pins the stress is so much below the maximum that it is improbable that the actual stress would exceed the 8420 maximum. At R it almost certainly would, however, and some means of reinforcing around this hole should be resorted to as has been indicated on *Fig.* 42.

The average stress at the right end is $2000/.5 \times 4 = 1000$ pounds per square inch. The departure from this figure is due to the curvature. The equations from *Fig.* 43*A* and 43*B* give the same stress at R. This will only be true when R passes through the center of curvature. Otherwise there will be a difference due to the tangential component of R.

This link has been discussed in detail here to explain the method of analysis of *Fig.* 39. Ordinarily inspection would indicate that $\theta = 90$ degrees in *Fig.* 43*B* gives the maximum stress, and this is all

that would need to be considered.

A curved link with eccentric radii is illustrated in *Fig.* 44. While the theory does not strictly apply to such a shape, it may be used to obtain a close approximation. The assumption is made that changes in contour at remote points will not affect the stress as long as the section and method of loading is not altered. The stress at the inside of the crotch, for $\theta = 90$ degrees, is found by increasing the outside radius the amount of the eccentricity, leaving the .819 dimension unchanged, as shown in *Fig.* 45*A*. Similarly for the outside edge, the inside radius is reduced to make it concentric as shown in *Fig.* 45*B*. These changes do not alter the section, nor the eccentricity of the load with regard to the center of the section.

The calculations for the stress are

$$\frac{6P}{bd^2} = \frac{6 \times 1475}{.25 \times .819^2} = 52700$$

In *Fig.* 45*A*, $r_o = (.469 + 1.288)/2 = .878$, $e = 1.91$, $k = .469/1.288 = .364$. From *Fig.* 39, $K_1 = 1.47$, $K_3 = 1.70$ and the stress at r_1 for $\theta = 90$ degrees

$$S_\theta = 52700\,(-1.47 \times .191 + 1.7 \times .878) = +63900$$

In *Fig.* 45*B*, $r_o = (.278 + 1.097)/2 = .687$, $e = 0$, $k = .278/1.097 = .254$. From *Fig.* 39, $K_4 = .53$ and the stress at r_2 for $\theta = 90$ degrees

$$S_\theta = -52700\,(0 + .53 \times .687) = -19150$$

In *Fig.* 46 is shown the photoelastic picture[3] of this link. $H = 86.0$, model load $= 61.3$ pounds.

$$At\ r_1,\quad S_\theta = \frac{7.5 \times 86 \times 1475}{.25 \times 61.3} = +62000$$

$$At\ r_2,\quad S_\theta = \frac{-2.5 \times 86 \times 1475}{.25 \times 61.3} = -20700$$

As a final example consider the *C* clamp of *Fig.* 47. Load is applied by the 1¼ inch diameter screw and is estimated at 5000 pounds. Maximum stress is obviously in the 7/16 inch fillet and would be expected somewhere near the 45 degree line. Here the outside radius is eccentric in two directions. Leaving the inside contour fixed and altering the outside to suit gives $d = 1.91$ inches, $r_1 = .438$, $r_2 = r_1 + d = 2.35$, $r_o = 1.39$, $k = .188$, $P = 5000$ pounds, $e = -1.062$. From *Fig.* 39, $K_1 = 1.93$ and $K_3 = 2.48$. The stress is then

$$\frac{6 \times 5000}{2.5(1.91)^2}[-1.93\,(-1.062) + 2.48 \times 1.39\,\sin 45^\circ]$$

$$= +14750 \text{ pounds per square inch}$$

This piece was checked by photoelastic analysis on ½ size model, giving plus 13,300 pounds per square inch, or 10 per cent below the elastic ap-

[3] This is Fig. 21 of Part IV of the article on Photoelastic Analysis, MACHINE DESIGN, June, 1940.

(*Concluded from Page* 138)

proximation. In this connection it is worth noting that the model was loaded unusually high so that some stress adjustment may have occurred. This would account for some of the 10 per cent loss. The writer believes that the method of approximation outlined here will give accuracy (say within 15 per cent), and be more satisfactory than the usual $P/A + Mc/I$ method. Photoelastic analysis is, of course, much preferred for such wide departures from the theoretical problem as this C clamp.

The above detailed discussion of specific problems shows the wide applicability of this method of analysis to a curved beam. It should be of material assistance to the machine designer, hard pressed as he is today, in securing more efficient disposition of material without an unreasonable expenditure of time.

Meetings and Expositions

June 9-11—

International Association of Blue Print and Allied Industries. Annual exposition to be held at French Lick Springs Hotel, French Lick, Ind. George G. Merry, 220 White building, Buffalo, is secretary.

June 10-11—

Radio Manufacturers association. Annual meeting and exhibition to be held at Stevens Hotel, Chicago. Bond Geddes, 1317 F street, Northwest, Washington, is secretary.

June 12-14—

Eastern Photoelasticity conference. Thirteenth semi-annual meeting to be held at Cambridge, Mass., under the auspices of the department of mechanical engineering at Massachusetts Institute of Technology. Additional information may be obtained from W. M. Murray, Room 1-321, M.I.T., Cambridge, Mass., chairman.

June 16-20—

American Institute of Electrical Engineers. Annual sum-Pacific coast air conditioning meeting and exposition to be held at the Exposition auditorium, San Francisco. Additional information may be obtained from the International Exposition Co., manager of the exposition, Grand Central Palace, New York.

June 16-20—

American Society of Mechanical Engineers. Semiannual meeting to be held in Kansas City. C. E. Davies, 29 West Thirty-ninth street, New York, is secretary.

June 16-20—

American Institute of Electrical Engineers. Annual summer convention to be held at Toronto. H. H. Henline, 33 West Thirty-ninth street, New York, is national secretary.

June 20-21—

American Society of Mechanical Engineers. Applied Mechanics division to hold meeting at University of Pennsylvania, Philadelphia. C. E. Davies, 29 West Thirty-ninth street, New York, is secretary.

June 23-26—

American Society of Agricultural Engineers. Annual meeting to be held at Andrew Johnson hotel, Knoxville. Raymond Olney, St. Joseph, Mich., is secretary.

June 23-25—

Institute of Radio Engineers. Semiannual meeting and exhibit to be held at Statler Hotel, Detroit. Harold P. Westman, 330 West Forty-second street, New York, is secretary.

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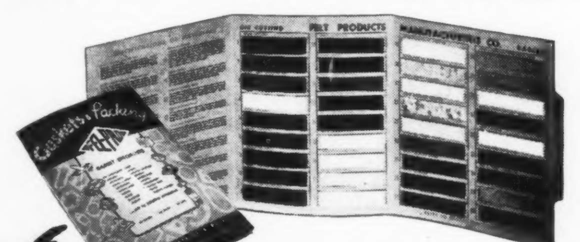

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Business and Sales Briefs

TO MAKE its general office organization an even more effective unit, Ralph M. Hoffman, vice president, has been delegated by Link-Belt Co., 307 North Michigan avenue, Chicago, to direct and supervise sales for the company. Mr. Hoffman, a mechanical engineer, has been assistant to the president since January 1940 and has been with the company since 1923.

♦

Transfer of W. T. Cooper, sales engineer, to Chicago, has been made by Bakelite Corp., New York. He will be located in the Carbide and Carbon building, 230 North Michigan avenue, Chicago. Mr. Cooper for the past four years has been associated with the company's molding material sales in the New York metropolitan area, and will continue in a similar capacity in the Chicago territory.

♦

According to an announcement by Aluminum Industries Inc., Cincinnati, operation has begun in the company's new $800,000 aluminum and magnesium foundry and aluminum paint plant. This is the first unit to be completed in the company's building program which will include, in addition to this plant, a large machine shop and office building. All of the new buildings are being erected on the site recently purchased by the company on Werk road, Cincinnati.

♦

For the past four years connected with sales promotion of motorized speed reducers manufactured by Janette Mfg. Co., 556 West Monroe street, Chicago, Max L. Robinson has been appointed sales manager of the company. Formerly sales manager of the Roth Brothers & Co. division of Century Electric Co., Mr. Robinson's sales experience in the electrical industry dates back to 1915.

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Taking over duties previously performed by V. H. Godfrey who has been recalled to active duty with the U. S. Navy, J. E. Skinner was recently placed in charge of welding wire sales as assistant to W. H. Bleecker, sales manager at the general sales office of Page Steel Wire division, American Chain & Cable Co. Inc., Monessen, Pa. W. G. Hoagland of the Chicago office of the Page division was transferred to Monessen to take over the duties previously handled by Mr. Skinner.

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With The Surface Combustion Corp. for the past six years, L. M. Lindsey has been named engineering sales manager of General Alloys Co., Boston.

♦

A. M. Wolf has been transferred to the St. Louis office of Cutler-Hammer Inc., Milwaukee. Mr. Wolf is a graduate electrical engineer from the Armour Institute of Technology.

♦

B. J. Brugge has been appointed welding consultant and engineer of Lincoln Electric Co., Cleveland, at Washington. He will be engaged in consulting work having to do with the application of arc welding in the National Defense program, and will be available for such work with all governmental de-

partments. He will be associated with the company's representatives, T. A. Canty Inc., Baltimore.

Among the recent appointments made by Laminated Shim Co. Inc., Glenbrook, Conn., is that of Richard Seipt as vice president in charge of sales.

Industrial Timer Corp. is now located at 117 Edison Place, Newark, N. J.

Formerly assistant general manager of sales, Pittsburgh Steel Co., Pittsburgh, J. K. Beeson has been elected vice president in charge of sales.

Lou R. O'Connor has been made representative for Ampco Metal Inc., Milwaukee, in the states of Washington, Oregon, Idaho and western Montana.

With headquarters in Cleveland Frederick A. Ohlmstead will act as district sales manager of the Youngstown Sheet & Tube Co., Youngstown.

In full charge of the activities of Universal Gear Corp., Indianapolis, in western New York, including the Buffalo and Syracuse areas, will be J. F. Johnson and associates, with offices at 707 Dun building, 110 Pearl street, Buffalo.

Appointment of Harry R. Brethen of Detroit as representative for the state of Michigan has been made by Erie Resistor Corp., Erie, Pa. Mr. Brethren will represent the plastic division which produces custom molded plastics by the injection and extrusion process.

General sales manager of Duraloy Co. since 1937, Harvey T. Harrison has been elected vice president in charge of sales. He joined Duraloy in 1928 in its New York office and three years later was made district manager at Cleveland, where he remained until 1937.

R. E. Spencer Geare is now connected with Thermoid Rubber division, Thermoid Co., Trenton, N. J. He is well known as an authority on V-belts and power transmissions, and has held sales and engineering executive positions with several prominent companies for 25 years.

Establishment of a new Metropolitan distributing branch with headquarters at 570 Lexington avenue, New York, is announced by the Great Electric Co. Earle Poorman, district manager of appliance sales in New York since 1931, has been appointed manager of the branch.

A member of the sales staff of Jessop Steel Co., Washington, Pa., for many years, William W. Britton has been made special representative for the company for the Pacific coast territory including California, Oregon and Washington. His headquarters are in the Chamber of Commerce building, Los Angeles.

G. Donald Spackman has been appointed general manager of Lukens Steel Co. and its subsidiaries, Coatesville, Pa. A graduate of Swarthmore college, with bachelor of arts and mechanical engineering de-

grees, Mr. Spackman joined Lukens Steel Co. in 1919 and was appointed general superintendent in 1936.

Robert L. Reed, president, Robert L. Reed Co., Detroit, has been elected president of the Plating and Rustproofing Association of America, Detroit. Charles Erdman, Central Plating Co., is vice president of the association.

Associated with Century Electric Co., St. Louis, since 1907, J. L. Woodress has been appointed director of sales. His first assignment with the company was that of traveling sales and service. He was successively assistant sales manager, sales manager, and general sales manager. Earl S. Moore succeeds Mr. Woodress as general sales manager, and C. E. White becomes export manager replacing Mr. Moore.

According to a recent announcement by Emil E. Hollander and Carl M. Peterson, president and treasurer-secretary respectively, Star Electric Motor Co., Bloomfield, N. J., is moving its offices and factory to a new building recently acquired. Moving has already begun and will be completed by January, 1942. The Star company was started thirty years ago, and manufactures electric motors, generators, geared-head motors, magnetic brakes and similar equipment.

A. H. Frauenthal, vice president and general manager for the past five years of Bantam Bearings Corp., South Bend, Ind., has resigned to establish a new company in Muskegon to manufacture large special roller and ball bearings. R. B. Nichols, who had been secretary and sales manager as well as assistant to Mr. Frauenthal at Bantam, will succeed him as vice president of Bantam Bearings Corp.

Ralph M. Hoffman has been elected a vice president of the Link Belt Co., Chicago, in charge of general sales.

Sherman Barnes has been placed in charge of the Western New York sales territory of Ampco Metal Inc., Milwaukee. He will make his headquarters at 239 Burr street, Rochester, N. Y. W. B. McKenzie, who previously handled this territory, is now at the company's Chicago office.

Well known in the industrial field, Roy S. Laird has been appointed sales manager for the Ohmite Mfg. Co., Chicago. Mr. Laird has been with the company five years and has the knowledge of the company's products and their applications.

Floor space of the Elastic Stop Nut Corp. plant at Union, N. J., has been doubled to meet increased demand for its line of self-locking nuts.

American Felt Co. has changed its address from 315 Fourth avenue, New York, to Glenville, Conn.

Completion of its third expansion in as many years to its plant at Erie, Pa., has been made by Lord Mfg. Co., the latest being the addition of 27,000 square feet of space. A factor behind this increase is the National Defense program which accounts for a substantial portion of the present orders.